Karl Th. Renius

Fundamentals of Tractor Design

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Baldham, Germany

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Dedicated to all persons who supported my professional career,
in particular Prof. Dr.-Ing. Dr.-Ing. E. h. Hans Jürgen Matthies (1921-2016) whose inspiring lectures at TH Braunschweig (about 1963) convinced me to direct my interest to agricultural engineering (as one important field of mechanical engineering). He offered me a position at his institute, became my doctoral adviser, my engaged mentor, my dear revered colleague and finally a fatherly good friend.

The author, riding his Fordson F (produced in 1924) at the Hohenheim Field Day 2009. He donated this tractor to the German Agricultural Museum at the University of Hohenheim, Germany. Photo courtesy *A. Zweygarth*

Univ.-Prof. Dr.-Ing. Dr. h.c. Karl Theodor Renius

1938 Born on the Renius Farm "Albertshof", Oranienburg, Germany
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1966-73 Assistant at Institute of Agric. Machinery, TU Braunschweig, active in teaching, research and industry projects
1973 Ph.D. at TU Braunschweig (Matthies/Vogelpohl) in hydraulics
1973-81 Manager of Advanced Engineering of Deutz tractors at the new KHD Development Center Köln-Porz
1976-82 Honorary Lecturer of tractor design at RWTH Aachen
1981 Manager of Deutz Tractor Development, KHD Köln-Porz.
1982-2003 Full Professor and Head of Institute of Agricultural Machinery at TUM, Technical University of Munich
2003 (retired) Honorary Head of Section "Mobile Machinery" at Institute of Automotive Engineering, TUM, Campus Garching

Main areas of professional expertise

Agricultural mechanization and tractor design. Drivetrains, hydraulics and terramechanics of tractors and mobile machinery, cumulative damage in fatigue, gobal product planning and technology management, education and technology transfer, ISO standards

Important national and international honorary positions and awards

Preface

The expected increase in the population of Earth will require an adequate increase in plant production, probably not just in proportion to population growth, but even greater due to expected higher meat consumption worldwide and anticipated higher demand for bio-energy and raw materials.

One key discipline to meet this target is agricultural mechanization.

It increases agricultural productivity in a dramatic manner as demonstrated in all industrialized countries in which the output of one agricultural worker has been raised, for example, by a factor of 100 or more; in some cases, such as tillage with large tractors or harvesting with self-propelled machines, by factors of between about 1000 and 5000.

Important additional benefits can be seen in constituting the precondition for *developing a nation from poverty to prosperity* due to a reduction in the number of agricultural workers and the transfer of the labour force to other sectors of the national economy.

This aspect is an ongoing important strategic subject for the *FAO*, *World Bank, Engineering Societies* and other institutions such as the *Club of Bologna*. In fact, most countries of the world are behind the industrialized ones in terms of economic performance. People leave their home countries not only because of natural catastrophes or wars but also due to poor living conditions. Hence, local improvements are urgent.

For this reason, this book addresses engineering fundamentals not only for the high technologies of industrialized countries, but also for less developed countries.

Consequently, proved basic designs and classical publications are included within defined technology levels, a principle which was first presented by the author in the CIGR Handbook of Agricultural Engineering, Vol. III, published by ASAE in 1999.

The roots of this new book at hand date to a Diploma Course held at RWTH Aachen University (1976-82) and since 1982 at the Technical University of Munich (TUM).

Participation in international conferences, working on international tractor projects, and contributing to ISO standards supported the international view. As a co-founder and Board Member of the *Club of Bologna*, the author benefitted for some 30 years from the worldwide networking of this unique group of leading agricultural engineers.

The book addresses not only students and practical tractor engineers but also colleagues of research, education, public tests, and engineering management. Facing the strong trend to global business it attempts to consider all ISO standards of agricultural tractors.

I would like to thank all colleagues of our community who supported this project.
My deepest gratitude goes to my wife Maria for her tireless support over the years.

Finally, my sincere thanks are also addressed to Stacy A. Marinelli for her patient corrections of my English and also to the Springer team for all its support.

Any suggestions for further improvement are welcomed.

Munich, 2019

Karl Theodor Renius
TUM, Technical University of Munich, Germany

Foreword I

Feeding the world is becoming an increasingly growing issue. The world is expected to carry up to 10 billion people by the year 2050. Thus, farm output has to double using more or less the same amount of farmland globally.

Obviously, agricultural mechanization is a main pillar for feeding the world, now and probably even more so in the future. In addition, agricultural mechanization is typically needed as the first step of general mechanization of a nation for its advancement from poverty to welfare and prosperity.

Global annual sales of tractors are estimated to be about 1.8 m units at 45 bn US$ (2017) sales volume. With that, tractors represent by far the largest segment in the agricultural machine industry and they play a central role in the mechanization of farming.

We forecast that this will stay unchanged in the next decades ahead of us.

However, the tractor in its current form will change. Interestingly, the verb “to tractor” does not exist, while the tractor performs all key jobs in the field, such as planting, plowing, mowing, and so on. The tractor will become more automated and autonomous, and at the same time tractor concepts will vary.

The tractor is increasingly becoming part of a connected and digitalized production system that enables resource optimization and sustainable farming.

But all these smart farming systems will only work if the base machine, the tractor, is innovative, reliable, and durable. That is what this book addresses: the fundamentals of good tractor engineering, as a foundation for all that will come.

From a personal perspective, I received the first edition of the book TRAKTOREN as a Christmas gift from my brother in 1988, when I was a working university student at our Mannheim factory. I read it with enthusiasm, not knowing at the time that I would have the privilege to become a member of the tractor industry. Until today it has laid for me the foundation for my knowledge around tractors.

I remember a conversation with Professor Renius in Berlin on August 21, 2013, about the need for an updated international “tractor book.” This vision has now become reality and the result at hand mirrors the expertise of an author who has been working on tractors and their components for more than 50 years in research, teaching, engineering, standards, patents, and worldwide consultation.

I congratulate Professor Renius on this new English reference book for science, education, and industrial practice on tractor design fundamentals and I am sure that I can thank him on behalf of so many others in our industry for having put together a work that is fundamental for all who study the technology in and around tractors in more detail for the years to come.

Mannheim, 2019

Markwart von Pentz

President Agricultural and Turf Division
Deere & Co, Moline, Illinois, USA

Foreword II

For more than 100 years, the tractor has been *the* key machine of agricultural mechanization. The launch of the “Fordson” tractor in 1917 – the first tractor with block chassis design and large scale production – boosted productivity and working comfort in the subsequent years. In the following decades, the tractor developed into a multi-purpose and mobile energy source with standardized interfaces to connect to a tremendous variety of agricultural implements. In the future, digitization will further advance automated and autonomous processes. Through connectivity, the tractor becomes an integrated part of the agricultural production system and “Precision Farming”.

The challenges in the future are not only in the economic and technological area. Especially environmental and social aspects have to be considered as well, on the one hand to produce sufficient food and renewable raw materials and on the other hand to ensure sustainable farming.

In this book, you will find the fundamentals of tractor technology, which have not really changed, but been enhanced over the years in a multitude of ways. An end to this trend is not in sight. Primarily the technology and layout of transaxles and hydraulics, combined with fatigue strength are covered in deep detail. Besides the engine, the understanding of these technologies is mandatory for an optimal tractor design. The content deals with the global requirement from base tractors with Low Tech up to High Tech versions.

The author of this book has played a major role in my career. He hired me to work at his Institute of Agricultural Machinery at the Technical University of Munich; he taught me tractor technology and inspired my enthusiasm for tractors. During this time, I had the privilege of working on the first edition of his German book TRAKTOREN. From this experience I know very well what it means to do a project like this. The first book has been with me throughout my time in the industry. After nearly 30 years in the tractor industry and nearly 20 years of responsibility for tractor engineering, I wholeheartedly supported an updated English tractor book on fundamentals with inputs and material from the industry.

This new English edition of tractor design will be invaluably useful for agricultural engineering, but in particular for the whole tractor community. With this book, knowledge and many years of experience on tractor technology will be maintained for present and future generations. It will be a set book for everyone dealing with tractors in one way or another. There is nothing better.

I congratulate Prof. Renius on this new international book and I am very thankful he made the effort to write it.

I wish him and his book all the best and great success.

Marktoberdorf, 2019

Heribert Reiter
VP Engineering Tractors Global, AGCO Corp., Duluth, Georgia, USA
Managing Director R&D, AGCO GmbH (Fendt)

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1. Mission, history, trends, markets, and costs

1.1 Global mission of agricultural mechanization

The prosperity of a nation needs mechanization in general [1.1]. However, a first step should address agriculture [1.2]:

1. The *classic role*: Mechanization of plant and animal production, storage, and processing in order to feed the continuously growing world population.
2. The *environmental role*: Mechanization of raw material and energy production and landscape maintenance in order to safeguard the planet.
3. The *strategic role*: Mechanization to increase labour productivity in agriculture in order to develop the prosperity of complete national economies.

The classic role. Mechanization supports the following two classic productivity factors of rural agriculture:

Land productivity and *labour productivity*.

Mechanization supports *land productivity* as an important factor among all others:

- *Breeding*	- *Plant protection*
- *Fertilizing*	- *Post-harvesting*
- *Irrigation*	- *Mechanization*

Mechanization benefits are, for example, improved yields, higher quality levels, reduced input materials, reduced human work loads, and reduced yield losses. The old methods of harvesting grain often have losses of 20 or even 30 % while the losses of a modern combine are only 1–2 %.

Influence on *labour productivity* is considerably higher – mechanization is even *by far the dominating factor*. A large tractor can plow more than 1,000 hand workers can within the same period of time.

The truly dramatic savings in labour cost from mechanization result in low world market prices and very high food quality. People in countries with highly mechanized agriculture spend only about 15 % of their income on food and beverages.

The environmental role requires adjusted machinery – in some cases completely new concepts and systems as well – but, in any case, tractors.

The strategic role of agricultural mechanization addresses the whole economy, as it can usually reduce the number of people working in agriculture. This leads to a transfer of manpower to other areas of the national economy, creating additional added value, as demonstrated for the US economy by **Fig. 1.1**. Under 2 % of people working in agriculture now (2019) produce even much more than ever before.

Remark: The bandwidth of the industrial sector can be broader in countries with high export volumes such as Germany.

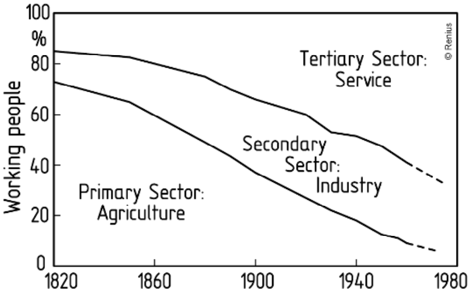

Fig. 1.1: Economic development of the United States by sectors [1.2]. Data from *Fourastié* [1.1]

Agricultural mechanization is usually the first technology step to overcome poverty, low life expectancy, high illiteracy, high infant mortality, a low level of infrastructure, a low level of social welfare, etc.

Regarding all machinery being in use, the tractor is by far the most important piece.

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1.2 Mission and use of the agricultural tractor

The **definition of the tractor** is, for example, settled by ISO 12 934:

> *The agricultural tractor is a self propelled agricultural vehicle having at least two axles and wheels or endless tracks, particularly designed to pull agricultural trailers and pull, push, carry and operate implements used for agricultural work (including forestry work) which may be provided with a detachable loading platform.*

One-axle tractors (also "walking tractors") [1.3, 1.4] are often a first step of mechanization; however, excluded from this book.

General importance of the tractor. The agricultural tractor is usually the most important piece of agricultural mechanization. During its first introduction in a country, sales volume among all agricultural machinery is typically 50 % or even more. "Even more" when tractors are not only used for agricultural production, e. g. also for non-agricultural transportation.

When the level of mechanization becomes higher and higher, the introduction of self propelled machinery (often beginning with combines) can lead to a reduction of the tractor sales in relation to the total sales value – for example to about 35 %. At the same time, non-agricultural transportation can become less important.

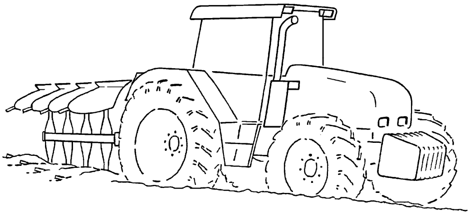


Fig. 1.2: Tractor use in Western Europe: Plowing is still important. Drawing: Author

Role of one-axle tractors. Standard tractors with two axles are relatively expensive for a first step of mechanization. One-axle tractors with rated engine power between 2 and 12 kW are often sufficient, mainly for small farms. This was – for example – the first step in many Asian countries such as Japan, Thailand, or China. Individual design is classified by power levels and application for dry land (upland) or paddy fields (mainly addressing rice production).

The production and use of one-axle tractors is still very important in China, but the general trend is to reduce the production in favour of two-axle tractors.

Typical tractor operations. There are many types of operations worldwide from country to country with significant differences of importance. As a general rule, the following groups can be defined:

1. Tillage
2. Sowing
3. Fertilizing
4. Plant protection
5. Irrigation (PTO)
6. Weeding
7. Harvesting
8. Front end loading
9. Transportation
10. Miscellaneous

Plowing is still a common operation in Western Europe, **Fig. 1.2**, usually with the largest tractor on the farm, while plowing in dry climates (also in North America) has become seldom.

The PTO (power take-off) is used for many types of operations (excluding transportation and front end loading): In Europe, the PTO is used for 30–40 % of the total working time, in Asia often more.

The total time of on-the-road driving (also with implements) takes about 1/3 of the total operating time in Europe. In developing countries mostly even more.

Typical power densities (kW/ha) for intensive farming are recommended in [1.5]:

- For small farms ≤ 31 ha: ..... ≥ 3.2 kW/ha
- For farms 31–195 ha: ...... 3,2–1.6 kW/ha
- For large farms ≥195 ha: ..... ≤ 1.6 kW/ha

1.3 Tractor history

1.3.1 Tractors with rear-wheel drive

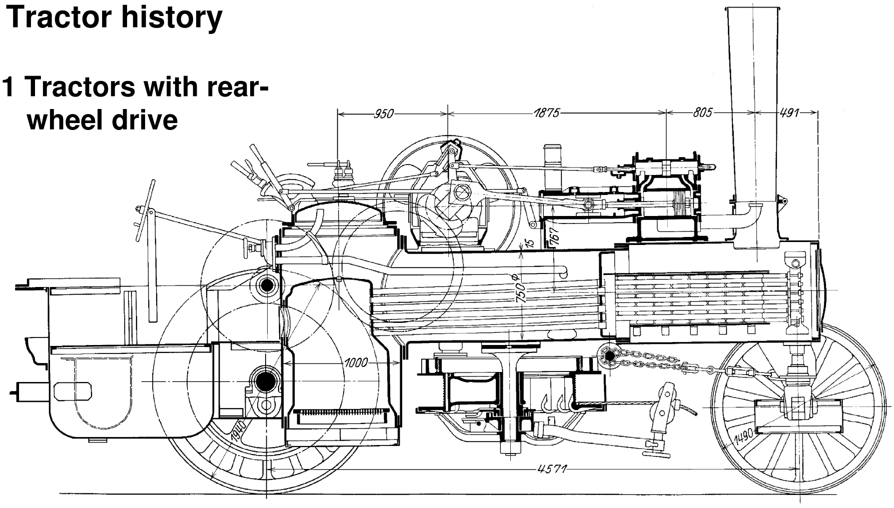


Fig. 1.3: Mobile steam engine of BORSG, 1920. Plowing by cable, mostly with two mobile units. Nominal power 81 kW at 15 bar and 320 °C steam [1.16], weight about 15 t. First units around 1860 by *John Fowler* (1826–64), UK, supported by *Max Eyth*, Germany (1836 –1906)

Mobile steam engines, Fig. 1.3, can be regarded as the forerunners of **steam traction engines** as introduced by J. I. Case in 1878 and steered by horses; self propelled in 1884. They were produced by Case until 1924.

In 1876, *N.A. Otto's* 4-cycle Internal Combustion (IC) engine reduced weights and costs [1.6–1.8], **Table 1.1**.

A very early gasoline tractor for which a practical agricultural application (in 1892) is documented is a prototype from *John Froelich*, USA, [1.9–1.11].

Hart-Parr created in 1902 one of the first tractors replacing steam power by an IC engine [1.8], production from 1904, **Fig. 1.4**. *Daniel Albone* presented his “Ivel Agricultural Motor”, a three-wheeler, in 1902 in England [1.12]. A further early tractor was the German 25 hp Deutz tractor, **Fig. 1.5**.

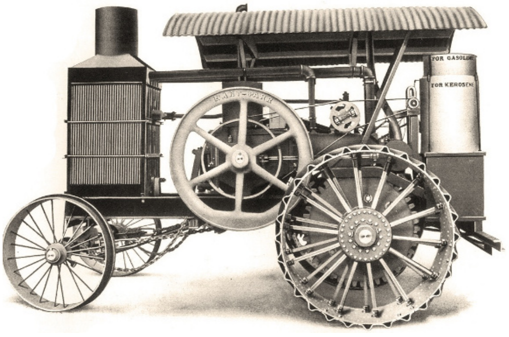


Fig. 1.4: Hart-Parr type 30-60 (60 hp) 1907–18. First production of Hart-Parr tractors: 1904 [1.8]

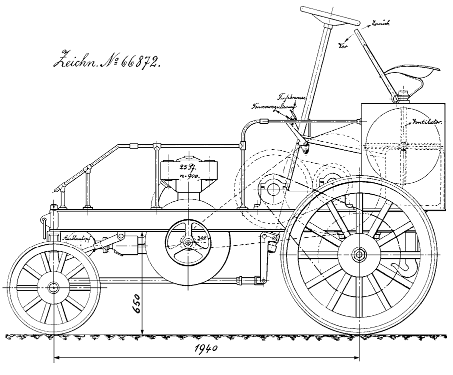


Fig. 1.5 (right)**:** Deutz tractor from 1907, prototypes. Courtesy Deutz

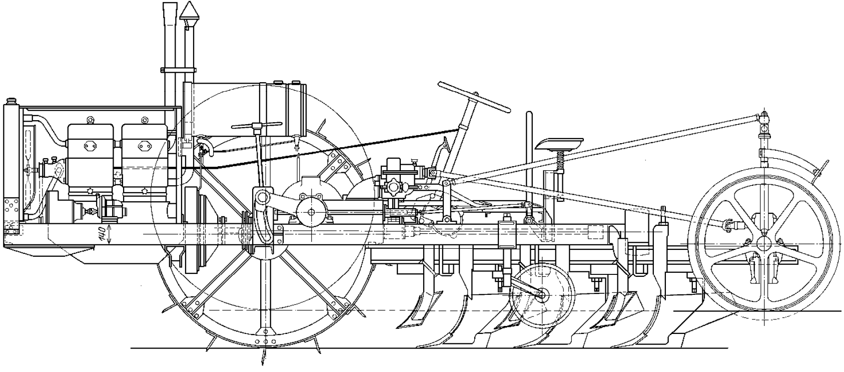


Fig. 1.6: Stock motor plow “Stokraft” 1911, 40 hp, 4-cylinder engine, weight 3,000 kg, 2,400 kg axle load. Wheel diameter 1.5 m [1.16]. Drive of stationary threshing machines by pulley

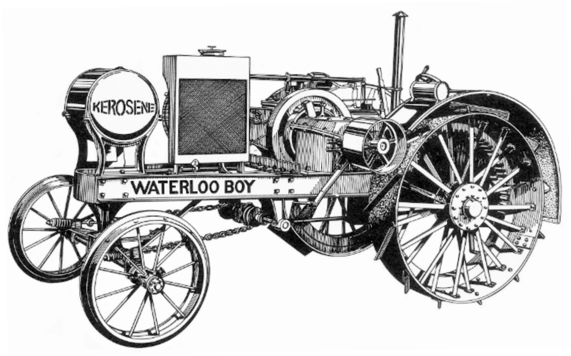


Fig. 1.7: The “Waterloo Boy”, Series N, 25 belt hp at 750 rpm, 2-cyl. engine, two forward speeds. See also Nebraska test No. 1 (1920). Courtesy John Deere

An **intermediate era of motor plows** was typical for Europe from 1910 to about 1925, **Fig. 1.6**.

An early important standard tractor was the “Waterloo Boy” (USA), **Fig. 1.7**. Production started in 1914 with models “L” and “LA”, followed by series “R” (1915–19) and “N” (1917–24).

The “Waterloo Gasoline Tractor Company” was purchased by John Deere in 1918 [1.9–1.11].

The 1920s were dominated by *Henry Ford’s* Fordson F, **Fig. 1.8**. 747,500 were produced [1.13].

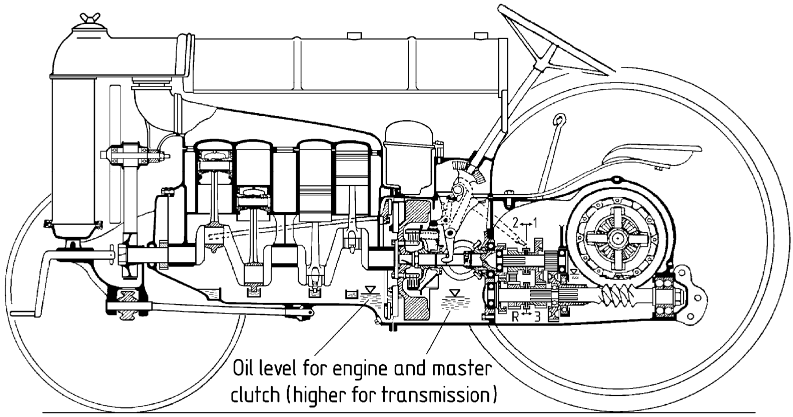

2↔1
R↔3
Oil level for engine and master clutch (higher for transmission)

Fig. 1.8: Fordson F (1917–28), first tractor mass-produced on assembly line. 20 hp at 1,300 rpm, kerosene. Net weight about 1,300 kg; had a pulley, but no rear PTO [1.13]. Several sources

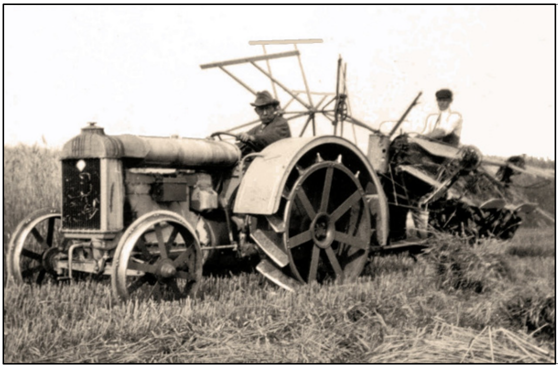


Fig. 1.9: Fordson F on the *Renius* farm “Albertshof” near Oranienburg, about 1926. Photo *W. Renius*

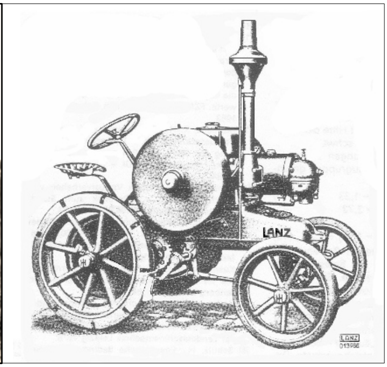


Fig. 1.10: 12 hp LANZ Bulldog 1921. 2-cycle “heavy oil hot bulb engine”, 1 speed [1.15]

A frameless chassis with a closed transmission had been introduced already in 1914 by the “Wallis Cub” tractor [1.14]. However, the block design was perfectly realized with the Fordson F; the low price was achieved by the high maturity of the design and the first tractor assembly line.

The Fordson F was proved by Nebraska Tests Nos. 18 (1920) and 124 (1926). The father of the author bought one, **Fig. 1.9**.

Competition arose in Germany from the LANZ Bulldog. The first design was extremely simple, **Fig. 1.10**; but achieved popularity for driving threshing machines. Later on, a considerably improved design was mass-produced (see the following page).

In 1923, the “McCormick Deering 10-20” from International Harvester, USA, opened a new era of remarkable tractor designs, **Fig. 1.11**.

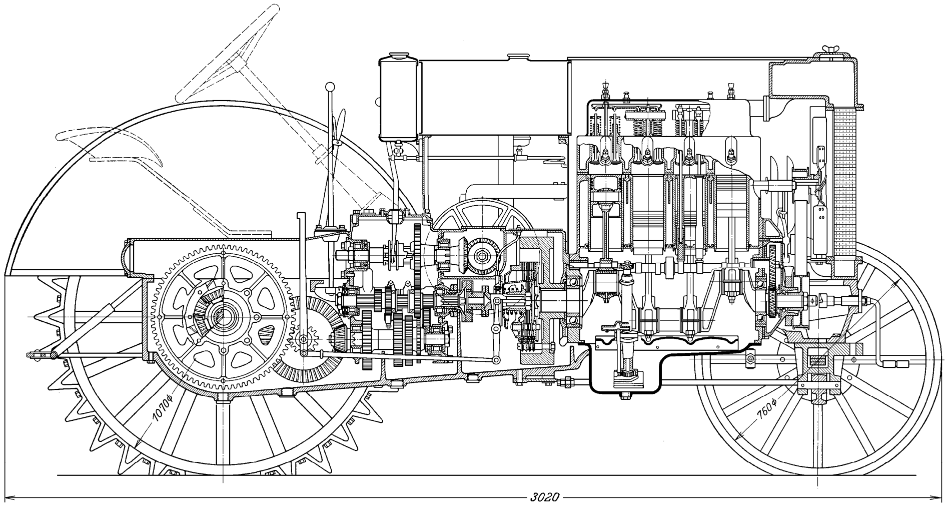


Fig. 1.11: IH “McCormick-Deering 10-20” (1923), 20 PTO hp, 3 speeds, pulley and PTO (not visible). First Nebraska Test was No. 95, 1923. Courtesy International Harvester 1930 [1.16]

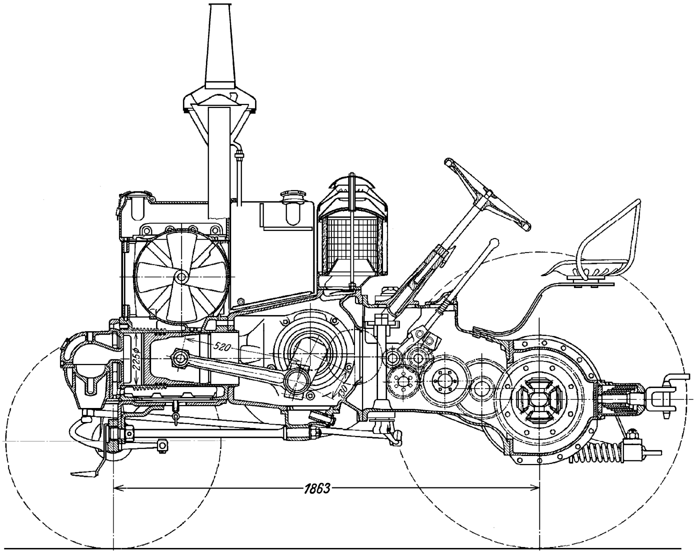


Fig. 1.12: “Kühlerbulldog” HR5 of LANZ Company 1929, improved cooling system. 30 hp at 500 rpm. Weight 2,560 kg [1.16]. A total of 219,253 bulldogs were produced between 1921 and 1960. John Deere purchased the LANZ AG majority in 1956, changed the name to “John Deere Lanz AG” and in 1967 to “John Deere Werke Mannheim” [1.9, 1.10], still a large plant (2019).

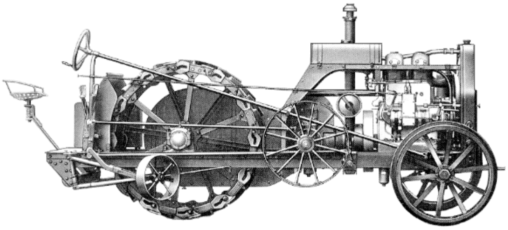


Fig. 1.13: First 4-cycle diesel engine in a vehicle: Tractor Benz-Sendling S6 (Prototypes 1922), 27 hp MWM engine, one driving wheel [1.18]

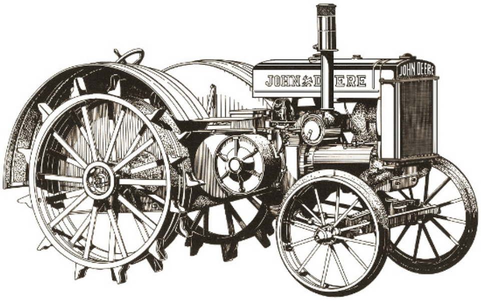


Fig. 1.14: John Deere model D 15-27, 27 hp, (kerosene) 1923 [1.9]. Courtesy John Deere

After six LANZ HR1 prototypes, series production of the HR2 Bulldog standard tractor started in 1926 on the first European tractor assembly line. LANZ chief engineer *Fritz Huber* (1881–1942): *“A tractor engine cannot be one- cylindrical enough.”* **Fig. 1.12** shows the improved “Kühlerbulldog” of 1929 [1.15–1.17].

The first *vehicle* with a 4-cycle diesel engine was the Benz Sendling S6 in 1922, **Fig. 1.13** [1.18, 1.19]. Series production from 1923 resulted in 100 units. Engine from MWM, (*Prosper l’Orange,* 1876–1939). Deutz produced its first diesel tractor MTH 222 (2 cylinder, 14 hp) in 1927.

John Deere introduced its very famous model “D” in 1923, **Fig. 1.14** [1.9], the basis for its rapidly increasing tractor business. The tradition of horizontal 2-cylinder engines (since the Waterloo Boy) was continued for about 30 more years. For details see Nebraska Test 102, carried out in 1924.

An outstanding person in tractor history was *Harry Ferguson* (1884–1960) [1.20]. He invented the three-point implement connection, British patents Nos. 160,248, 186,172, 195,421, 226,033, and 253,566.

He recognized that a low tractor mass means a lower price, but is not satisfying in the case of heavy pulling such as plowing.

His patent No. 253,566 from 1925 can be regarded as the "Ferguson master patent" – the original text from page 3 [1.21] is:

It is an object of the present invention to make it possible to reduce the weight of the tractors for agricultural work. A further object of the invention is to avoid or at least to reduce considerably the drag caused by the sinking of the implement into soft ground and to maintain the implement at a regular working depth in the ground.

The main progress was to control the working depth of implements by a closed loop control and at the same time to transfer vertical implement forces (weight and soil forces) to the tractor rear axle.

In his patent, he claimed three types of actuator power: *mechanical, electrical, and hydraulic.* The hydraulic version was first realized in series production in 1936 with David Brown, **Fig. 1.15**.

Henry Ford adapted the Ferguson System to his tractor "9N" (1939–47) by license through the famous "handshake agreement" of 1938. This became a real breakthrough with about 300,000 units sold.

The "*power take-off shaft*" (PTO) had been introduced by IHC (USA) in 1918, gaining importance with the "Farmall" [1.23], *also initiating the soon-to-be popular class of US row crop tractors,* **Fig. 1.16**.

Allis-Chalmers (AC) was the first company to replace steel wheels by air inflated rubber tires from 1932, **Fig. 1.17** [1.8].

The Allis-Chalmers tractor "WC" was the first one tested in Nebraska (No. 223, 1934) with both, iron wheels and rubber tires.

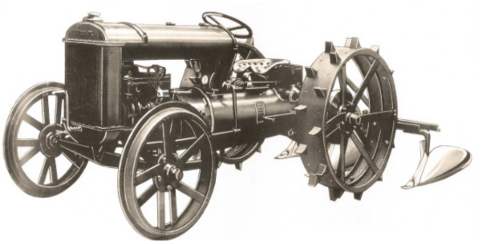


Fig. 1.15: The Ferguson-David Brown "Black Tractor". Prototype 1933, production 1936–39, 20 hp. It used the "Ferguson System" for the first time [1.8, 1.22], 1,250 were produced

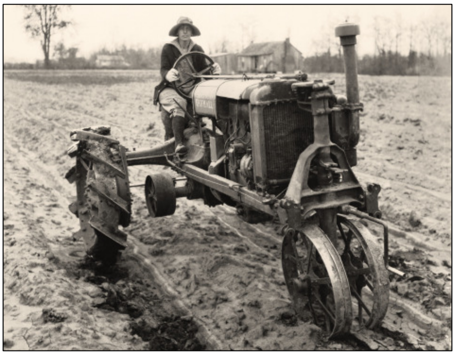


Fig. 1.16: McCormick-Deering "Farmall", 1924, 20 hp, early PTO, new row crop concept. Nebraska Test 117 (1925). Courtesy CNH

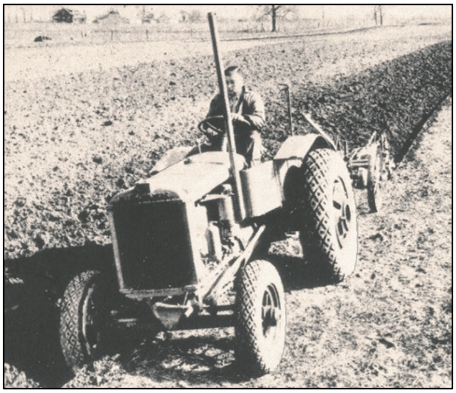


Fig. 1.17: Allis-Chalmers model U with air inflated rubber tires and a 3-bottom plow [1.8]. First experiments had been carried out with Firestone aircraft tires and reduced pressures

While only 14 % of all new tractors were fitted with pneumatic tires in the US in 1935 (at high extra costs!), the percentage was 95 % in 1940 – similarly in Germany, supported by Prof. *Helmut Meyer.*

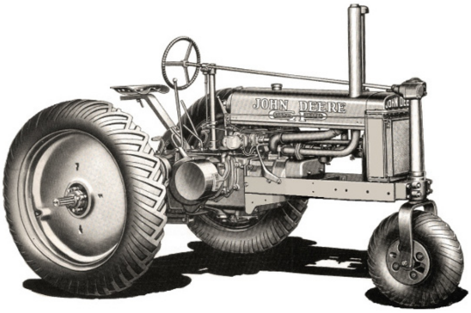


Fig. 1.18: John Deere model “BN”, version with pneumatic tires, 1935 [1.9]. Courtesy John Deere

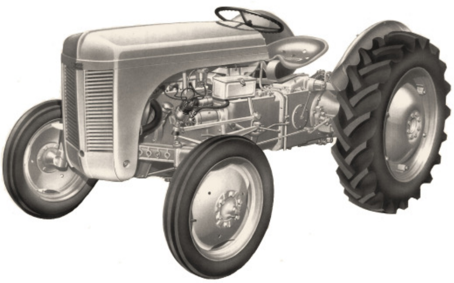


Fig. 1.19: Ferguson TE 20, 24 hp at 1,500 rpm with the US Continental Z-120 engine. 4 speeds forward. Production in Coventry, UK. 315 units in the first year 1946. Courtesy AGCO

The early John Deere model shown in **Fig. 1.18** is an example of US tricycle row crop tractors with their typical “overhead” steering which became very popular in the 1930s.

In the meantime *Harry Ferguson* developed his own tractor TE 20, **Fig. 1.19**. It was similar to the abovementioned Ford 9N, also applying the 3-point hitch and draft control, but its location of production changed to the UK (engine from US) [1.22].

This tractor initiated the worldwide introduction of the automatic draft control.

A typical German trend of agricultural mechanization from about 1935 to about 1955 was the introduction of low hp tractors in order to achieve a general fast replacement of horses. A popular example was the 11 hp Deutz, **Fig. 1.20** (1936).

Deutz (Klöckner-Humboldt-Deutz AG) became a leading tractor manufacturer in Europe after World War II, introducing air-cooled diesel engines for all its tractors from 1950, becoming the largest supplier of this concept worldwide for several decades.

However, the *very first* tractor available in series production with an air-cooled diesel engine was from the private German company Eicher in 1948, **Fig. 1.21**, with one cooling air blower for each cylinder.

Remark: Ref. [1.23]–[1.28] are cited in table 1.1.

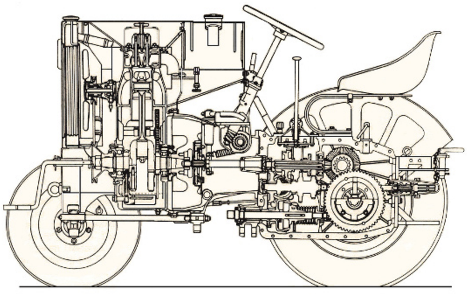


Fig. 1.20: German tractor Deutz F1M414, 1936, 1-cyl.-diesel engine, 11 hp, 3 speeds, mid and rear PTO. Courtesy Deutz

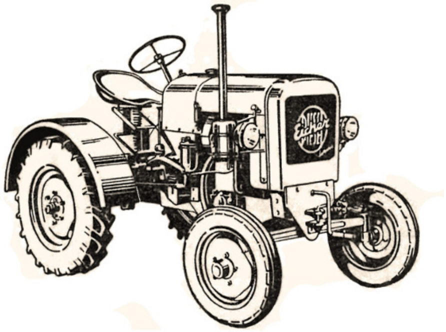


Fig. 1.21: First tractor produced in series with an air-cooled diesel engine: Eicher 16 hp, 1948 [1.16]. Source: Advertisements

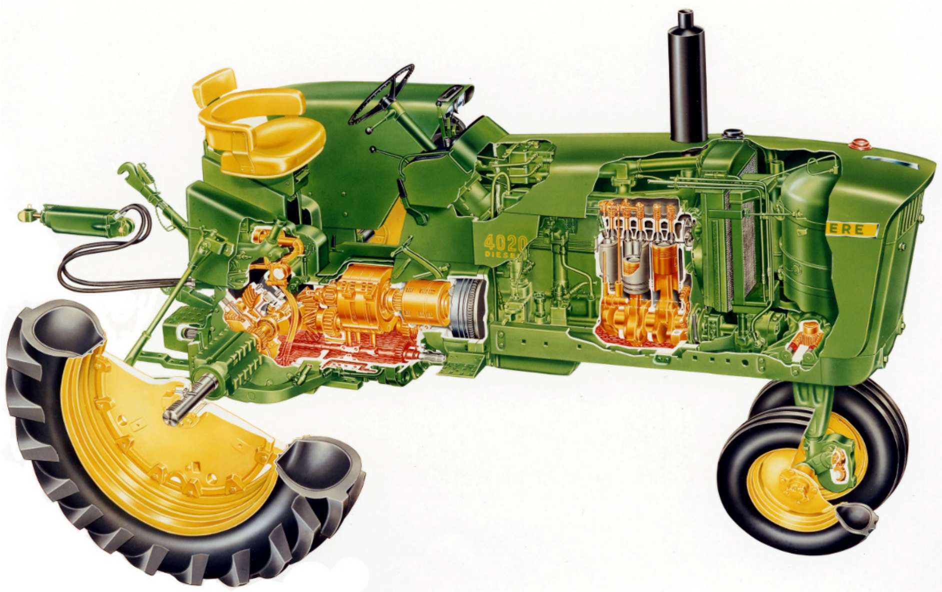
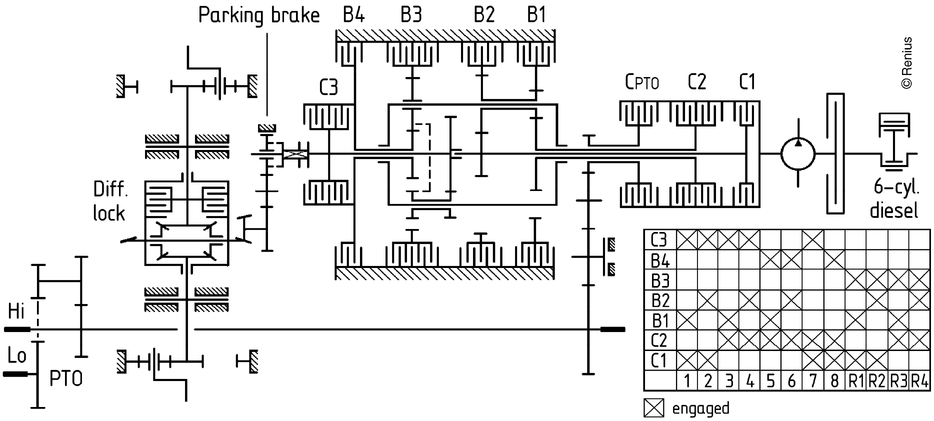


Fig. 1.22: John Deere 4020, 1963. 6-cyl. diesel engine, max. PTO power 91.2 hp at 2,200 rpm (Nebraska Test 849, 1963). Wheel base 2.48 m, rear tires 18.4-34, net weight with operator 8,945 lb, 8/4-speed full power shift (see below), constant pressure closed-center hydraulics with variable displacement pump, power steering, wet power brakes Courtesy John Deere

In 1960, John Deere presented under *William Hewitt* (1914–98) the tractor line 1010/2010/3010/4010 (36–84 PTO hp), ending the “only two-cylinder” era. Almost everything was new [1.10], also the constant pressure hydraulics. From 1963, the 10 series was updated and the 4020 became one of the most popular US tractors ever built, **Fig. 1.22**: It contained the “Power Shift” transmission as a popular version, **Fig. 1.23.**



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C2			☒	☒	☒	☒	☒	☒			☒	☒
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	1	2	3	4	5	6	7	8	R1	R2	R3	R4

☒ engaged

Fig. 1.23: “Power Shift” of John Deere 4020; eight well spaced speeds forward (about 2.8–29 km/h), and four reverse (about 3.3–9.3 km/h), both at 2,200 rpm rated engine speed. Excellent handling.

1.3.2 Tractors with all-wheel drive

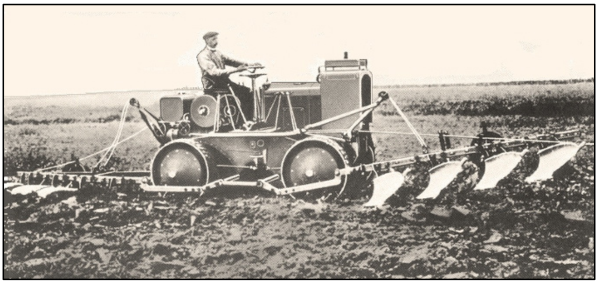


Fig. 1.24: "Deutzer Automobilpflug" 1907, 40 hp, auxiliary cable for lifting the mounted plows and as "traction aid". Patents of Brey and Heyer. Several prototypes. Courtesy Deutz

All-wheel drives (3WD, 4WD) were, for a long time, rather an exception [1.29]. The probably first 4WD drive of an agricultural tractor was realized by the Deutz "automobile plow" from 1907, **Fig. 1.24**.

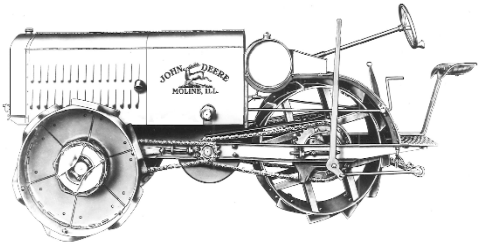


Fig. 1.25: "Dain" tractor, three driven wheels, 1915. 4-cylinder engine, 24 belt hp. 100 units by 1919. Courtesy John Deere

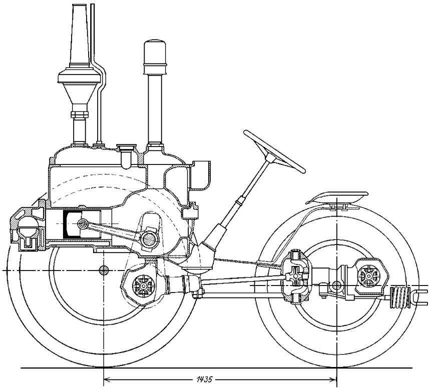


Fig. 1.26: LANZ Bulldog "HP", 4WD articulated, 15 hp, 1 speed, 723 units 1923–1926 [1.16]

Further 4WD milestones until 1950:

- 1912 "Nelson" tractor, 3 models, USA
- 1912 "Olmstead" tractor, USA
- 1915 "Dain" tractor, USA
- 1922 "Rogers" tractor, USA, articulated
- 1923 LANZ 4WD, articulated
- 1929 E 15/25 of Four-Wheel-Tractor Co., (Nebraska Test 165/1929)
- 1930 Massey Harris "General Purpose" (Nebraska Test 177/1930)
- 1949 UNIMOG
- 1949 MAN standard tractor
- 1950 SAME with front drive announced

The variety of designs was very high. In 1915, Joseph Dain Sr., in co-operation with John Deere, presented a tractor with three driven wheels, **Fig. 1.25**. Its field performance was evaluated to be very good:

"It pulled a 3-furrow plow with 14″ bottoms, working 6″ deep at 2 ½ mph" [1.9].

But the cost target was not met and a series of only 100 units was completed in 1919 by John Deere [1.9, 1.10].

The 4WD "Rogers Tractor" (65 hp, very early power steering) is said to be the origin of the large US articulated 4WD tractors (Nebraska Test No. 84, 1922). However, the LANZ Bulldog of **Fig. 1.26** was said (by John Deere) to be the first articulated 4WD tractor being produced in series.

4WD tractors with Ackermann steering offer the advantage of simple steering with lower forces than for articulated 4WD.

An early example was the model E 15/25 of the Four Drive Tractor Co., **Fig. 1.27**, USA, with four equal iron wheels.

More important was the Massey Harris “General Purpose” tractor with 4WD and Ackermann steering, **Fig. 1.28**. Its outstanding high axle clearance offered special benefits for row crops.

The small German company Boehringer developed the UNIMOG, **Fig. 1.29**.

This 4WD tractor became a legend although the beginning series production was not easy for the company and many farmers did not recognize its benefits for both field work and road transportation. The complete business was taken over by Daimler-Benz on Oct. 27, 1950.

See some more about this story on the following page.

Another progressive 4WD design was the MAN 25 hp tractor, **Fig. 1.30**, demonstrating the benefits of an additional front axle drive for standard tractors and those of the fuel-saving direct injection.

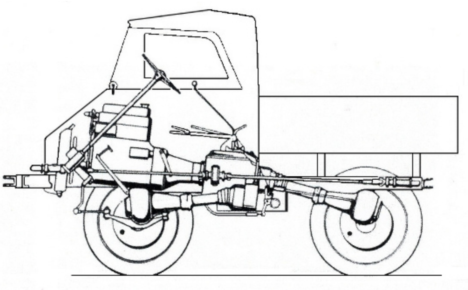


Fig. 1.29: UNIMOG structure. 25 hp, 50 km/h, fully suspended. Production from 1949 by Boehringer, from 1951 by Daimler-Benz in Gaggenau. Source: Patent drawing [1.18]

Fig. 1.30 (right)**:** MAN tractor with additional front drive and direct injection diesel engine, 25 hp, 1949. Source: Public advertisement

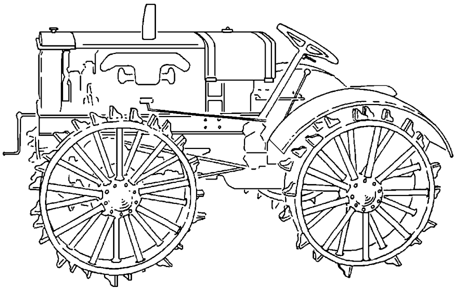


Fig. 1.27: 4WD Model E 15/25 of the Four Drive Tractor Co., USA. Ackermann steering [1.30], 30 hp, 2,950 kg, Nebraska Test No. 165 (1929). © Renius

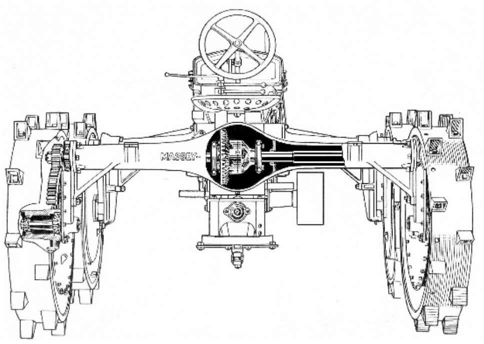
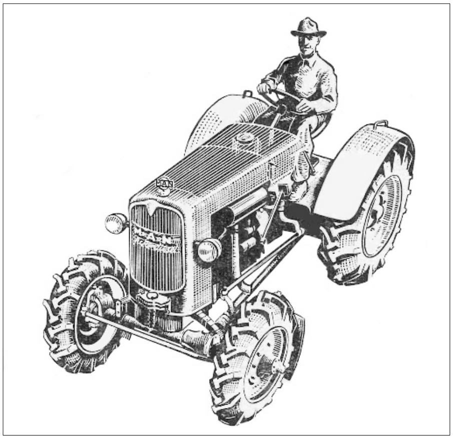


Fig. 1.28: Massey Harris “General Purpose” 4WD tractor 1930−38. Ackermann steering, net weight about 1,800 kg. Nom. power 22 belt hp. Nebraska Test No. 177 in 1930. The company has sold at least 8,000 units. Courtesy MF



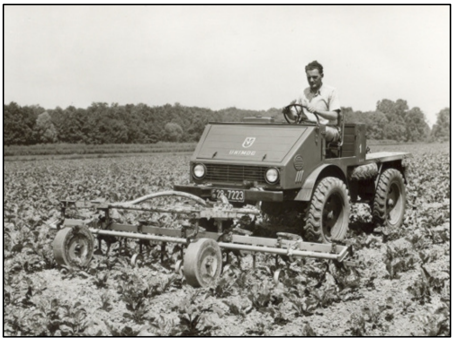


Fig. 1.31: First UNIMOG 1950 in a typical field operation. 25 hp. Advantages due to the low weight (only 1,680 kg) and the four-wheel drive which also enabled tillage, even plowing better than expected. Courtesy Daimler AG

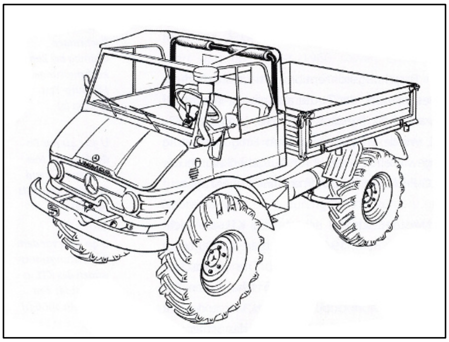


Fig. 1.32: UNIMOG, typical design of 1982. The high top speeds (initially 50 km/h, later more), the full suspension, and the large variety of functions opened other markets including military equipment. Courtesy Daimler AG

The UNIMOG was initially designed for an application only in agriculture, **Fig. 1.31**. Its performance for deep plowing was initially questioned, but finally much better than expected which can be explained by modern terramechanics (chapter 3.7). The UNIMOG became soon a universal vehicle also outside agriculture, **Fig. 1.32**.

In 1973, Deutz started with its INTRAC an interesting new front-cab tractor design (promoted by *A. Gego*) with integrated front hitch and front PTO. At the same time, Daimler-Benz introduced with the MB-trac a similar concept which was now more suitable for agriculture than the UNIMOG, for example due to larger wheels.

Continuously growing tractors approached the limits of traction and soil compaction. One solution are rubber tracks, first used by the Caterpillar Challenger 1987. Case IH came with its “Quadtrac” in 1997 with articulated steering, **Fig. 1.33**.

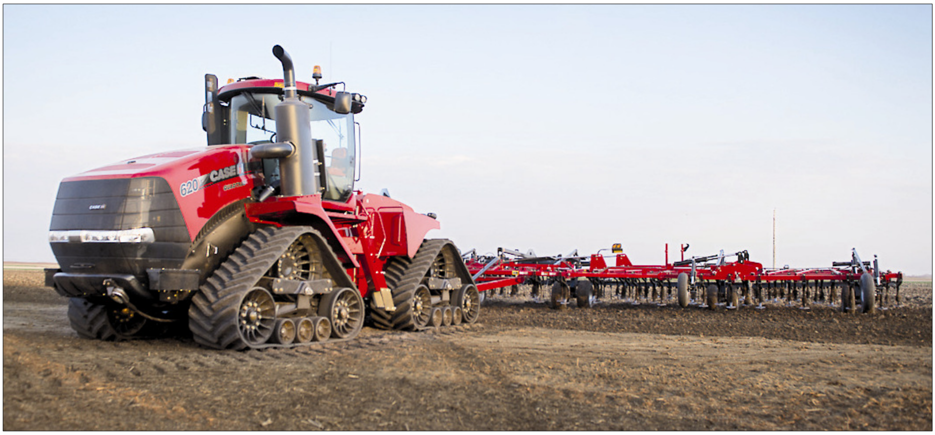


Fig. 1.33: Case IH “Quadtrac”, introduced in 1997 with about 265 kW rated engine power. This picture shows the later top model 620 as of 2013: 447 kW rated engine power (ECE R 120), net weight 22 t, diesel fuel tank 1,820 l plus AdBlue tank 250 l. Courtesy Case IH

Table 1.1: Milestones of tractor history

1876 4-cycle "Otto" engine, basic concept by *Nikolaus August Otto* (1832-91), Germany [1.6]

1878 Steam traction engine, Case, USA [1.23]

1889 Very first gasoline tractors, USA [1.23]

1890 Steam crawler tractor (B. Holt), USA [1.23]

1892 *Rudolf Diesel* (1858-1913) registers his German patent No. 67 207, accepted 1893

1892 *Froelich* gasoline tractor, USA. Field operation of one prototype reported [1.10, 1.23]

1902 Hart-Parr gasoline tractor, USA [1.8, 1.23]

1902 "Ivel" tractor (prototype), UK (Dan Albone)

1904 Crawler gasoline tractor (prototype) by *Benjamin Holt*, founder of Caterpillar [1.24]

1907 ASAE founded (SAE in 1905) – both USA

1908 Self propelled plow, Stock, Germany [1.16]

1914 "Waterloo Boy", USA (1918 J. Deere) [1.10]

1917-28 "Fordson F", block chassis, first tractor assembly line. About 735,000 units [1.13]

1918 First PTO by IH "8-16" (50 units) [1.23]

1920 Nebraska Test No. 1: Deere Waterloo Boy

1921 "LANZ-Bulldog," Germany (1 cyl., 2-cycle hot bulb engine 12 hp, 1 forw. speed *(Fritz Huber)*

1922 First diesel engine in a tractor (3 prototypes), Benz-Sendling, Germany. Series 1923

1922 "Rogers Tractor", articulated, hydraulic power steering (Nebraska Test 84/1922)

1924 McCormick-Deering "Farmall" (IH) [1.23]

1925 3-point hitch and draft control. Brit. patent No. 253 566 by *Harry Ferguson,* UK (1884-1960)

1927 First PTO standard issued by ASAE, USA

1933 Low pressure pneumatic tires at Allis-Chalmers USA) [1.8]. Nebraska Test 223 (1934)

1945 Life PTO. Cockshutt/Can., Model 30 tractor

1948 Air-cooled diesel engines by Eicher, Germany (from 1950 all Deutz tractors air-cooled)

1949 MAN 4WD tractor with direct injection diesel engine (Germany), other companies follow

1949 Daimler-Benz UNIMOG (25 hp, 50 km/h, 4WD, spring suspension, platform) [1.18]

1950 Fluid coupling, Voith/Allgaier-Porsche AP 17

1954 "Torque Amplifier", IH, USA (first Hi-Lo)

1954 Power steering by John Deere, USA [1.9]

1956 John Deere has majority of LANZ.

1958 Ford "Select-O-Speed": first full power shift transmission (10 forw./2 rev), USA/UK

1959 Pendulum test for new tractors, Sweden

1960 Kubota produces the first small Japanese two axle-tractor (Shibaura starts in 1961)

1960 Constant pressure hydraulics with variable displacement pump, J. Deere 3010 & 4010

1961 First tractors produced in India [1.25]

1963 J. Deere becomes No. 1 in the US tractor market due to the 4020 ("Power Shift" 8/4)

1965 "Central shaft" for front drives, SAME Italy

1966 First commercial hydrostatic tractor transmission: EICHER-Dowty, Germany-UK. 71 units

1967 IH 656, first mass produced tractor with hydrostatic transmission (USA), 6620 units

1970 Radial ply tyres for tractors by Kleber

1972 First quiet cab in Nebraska-Test No. 1110, J. Deere 4430, max. 82,5 dB(A) at driver's ear

1973 Deutz "INTRAC" and DB "MB-trac" in series, basic impulses for standard tractors

1973 Load-sensing hydraulics, Allis-Chalmers, USA, saves energy. No general breakthrough

1978 Electronic 3-point hitch control, Bosch

1980 40 km/h top speed for standard tractors, Fendt and Schlüter, Germany

1982 Compact driven 50 deg. front axles from ZF / J. Deere [1.26] and Deutz / Sige, Germany

1986 Kubota "Double Speed Turn", Japan

1987 Breakthrough of load-sensing hydraulics, initiated by Case-IH "Magnum," USA

1987 First soft cab suspension, Renault, France

1987 Caterpillar Challenger, rubber tracks, USA

1988 DIN 9684-1: German draft standard for tractor-implement communication [1.27]

1988 "Munich Research Tractor", ¾ frame, CVT

1990 Bosch-Fendt 3-point hitch vibration damping

1992 J. Deere ¾ frame tractors 6000 and 7000

1993 50 km/h for standard tractors presented, hydro front axle suspension, Fendt, Germany

1996 First hydrostatic power split CVT for standard tractors in series, Fendt "Vario 926"

1997 Case IH "Quadrac" with 4 rubber tracks

1997 Valtra "Sigma Power" engine boost [1.28]

1997 Beeline (Australia): First automatic steering

1999 GEO TEC (Germany): First auto navigation

2006 Fendt 936 (AGCO) with 60 km/h top speed

2008 John Deere 7430 and 7530 tractors with high voltage on-board net and el. remote power

2013 Factory-installed integrated tire inflation control system, Fendt (AGCO) 800 & 900

2015 Infinitely variable 4WD, Fendt 1000 line.

2019 Electrical power split CVT eAutoPowr of John Deere, announced at AGRITECHNICA

1.4 Recent technical trends and tractor concepts

Since about 1950, a remarkable technical tractor development has been typical for the industrialized countries.

In some areas such as Europe and Japan, additional front-wheel drive for standard tractors changed from an option to standard equipment [1.26, 1.29, 1.30], **Fig. 1.34**.

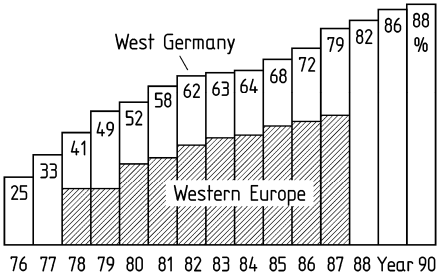


Fig. 1.34: Market shares for new tractors with both axles driven. Source: [1.30], modified

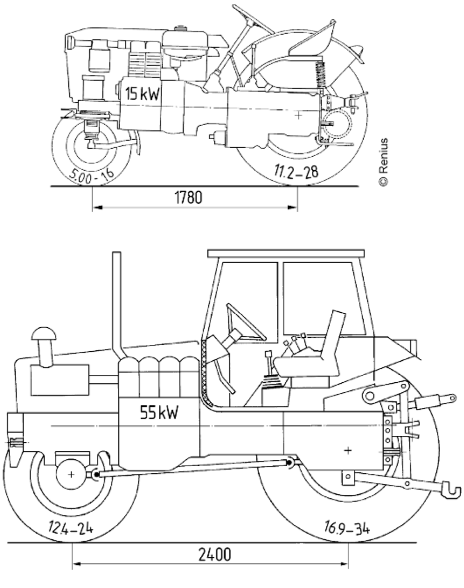


Fig. 1.35: Popular "average" tractor concepts in Western Europe 1958 (15 kW, above) and 1983 (55 kW, below). Same scale for both [1.31]

The same happened with closed quiet cabs containing integrated roll-over protective structures (ROPS), **Fig. 1.35**.

During the 1990s manufacturing systems basically changed from mass production volumes of large fixed lots to individual tractors being produced in a mix, each to customer-demanded specifications. This occurred along with a dramatic increase in functions and power levels.

Unusually high engineering efforts had to be undertaken in the developed countries since 2000 to reduce emissions to a passenger car level (EU & US regulations).

The most recent trends are characterized by the increasing role of information technologies (ISOBUS), improved powertrain concepts (power shifts, CVTs), better hydraulics (load sensing, proportional valves) and drive-by-wire systems.

Definitions relevant for EU homologations are listed in **Table 1.2**.

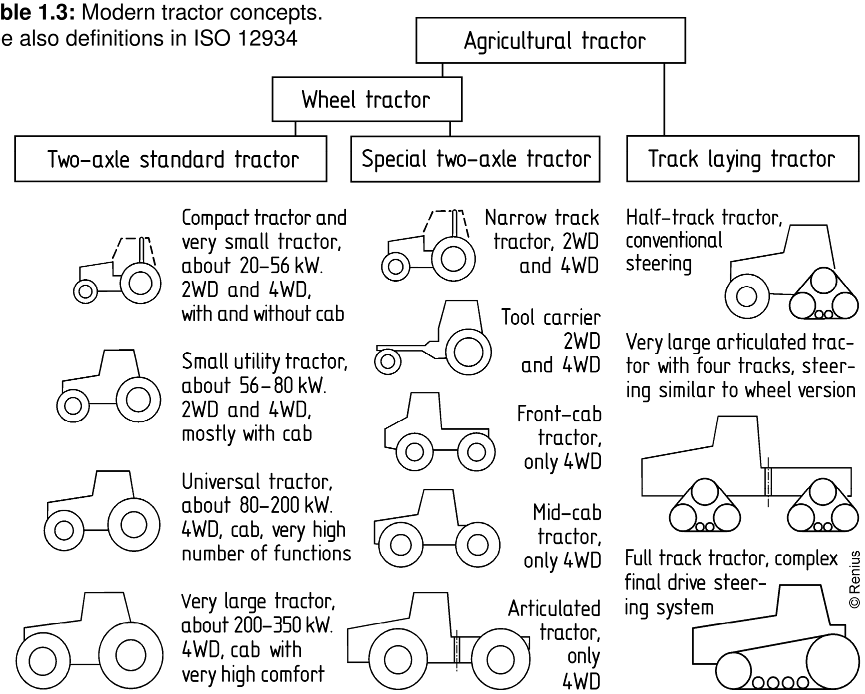
Table 1.2: EU Tractor categories

T1	Stand. tractors ≤ 40 km/h, track ≥ 1150 mm
T2	Narrow track tractors ≤ 40 km/h, track width ≤ 1150 mm
T3	Compact utility tractors ≤ 40 km/h, unloaded mass ≤ 600 kg
T4	Special purpose tractors
T4.1	High clearance tractors
T4.2	Extra-wide tractors
T4.3	Low-clearance tractors
T5	Tractors > 40 km/h top speed

A simplified "family tree" is shown in **Table 1.3**. The track laying principle was not important in the past decades but may gain some more attention in the future because of upcoming limits of tire dimensions for large tractors (chapter 3).

The **two-axle standard tractor** with about 50–200 engine kW (70–270 hp) dominates within the group of upper technology levels, **Fig. 1.36**.

Table 1.3: Modern tractor concepts. See also definitions in ISO 12934


Agricultural tractor
Wheel tractor
Two-axle standard tractor
Special two-axle tractor
Track laying tractor
Compact tractor and very small tractor, about 20–56 kW. 2WD and 4WD, with and without cab
Small utility tractor, about 56–80 kW. 2WD and 4WD, mostly with cab
Universal tractor, about 80–200 kW. 4WD, cab, very high number of functions
Very large tractor, about 200–350 kW. 4WD, cab with very high comfort
Narrow track tractor, 2WD and 4WD
Tool carrier 2WD and 4WD
Front-cab tractor, only 4WD
Mid-cab tractor, only 4WD
Articulated tractor, only 4WD
Half-track tractor, conventional steering
Very large articulated tractor with four tracks, steering similar to wheel version
Full track tractor, complex final drive steering system
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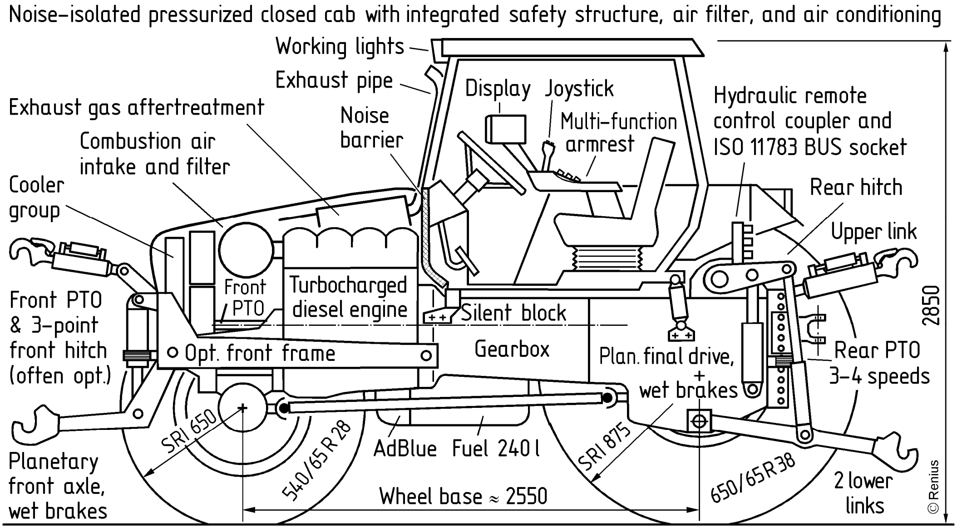
Noise-isolated pressurized closed cab with integrated safety structure, air filter, and air conditioning
Working lights
Exhaust pipe
Exhaust gas aftertreatment
Noise barrier
Display
Joystick
Multi-function armrest
Hydraulic remote control coupler and ISO 11783 BUS socket
Combustion air intake and filter
Cooler group
Rear hitch
Upper link
Front PTO
Turbocharged diesel engine
Silent block
Front PTO & 3-point front hitch (often opt.)
Opt. front frame
Gearbox
Plan. final drive, + wet brakes
Rear PTO 3–4 speeds
2850
SRI 650
540/65 R 28
AdBlue
Fuel 240 l
SRI 875
650/65 R 38
Planetary front axle, wet brakes
Wheel base ≈ 2550
2 lower links
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Fig. 1.36: Typical modern standard tractor, dimensions for about 110 kW max. PTO power. Typical options: Suspended front axle, soft suspended cab, front hitch, front PTO, ISOBUS, automatic guiding. Net weight (with front hitch and front PTO) about 6,500 kg, maximum weight about 11,000 kg

At the same time, the power range over 80 kW (110 hp) represents the most updated technologies.

Three popular tractor families as developed in Europe are presented in cutaway graphics.

Fig. 1.37 shows the John Deere 6R lineup, which belongs to the 6000 family, first introduced in 1992 in Mannheim (with a ¾ steel frame) and becoming one of the most successful tractor families ever built.

Fig. 1.38 presents the also popular New Holland T7 family of CNH Industrial with a widely common inner structure for NH, Case IH, and Steyr. A certain standardization of the powertrain for different brands can reduce investment costs for development and tooling; however, is not easy to handle for the sales management.

Fig. 1.39 shows the Fendt 700 Vario family of AGCO, which is very popular in Europe, mainly in Germany and France. The "Vario" technology of a continuously variable power split transmission (CVT) was developed by Fendt, introduced in series in 1996 and has initiated a new era of automatic CVT transmissions for tractors – now even with an impact on construction machinery.

Smaller tractors offer usually moderate technology levels with an exception: In Japan, a high technology level is achieved also for lower power levels according to the special structure of Japanese agriculture with predominantly small farm sizes. These tractors are also exported to many highly developed countries.

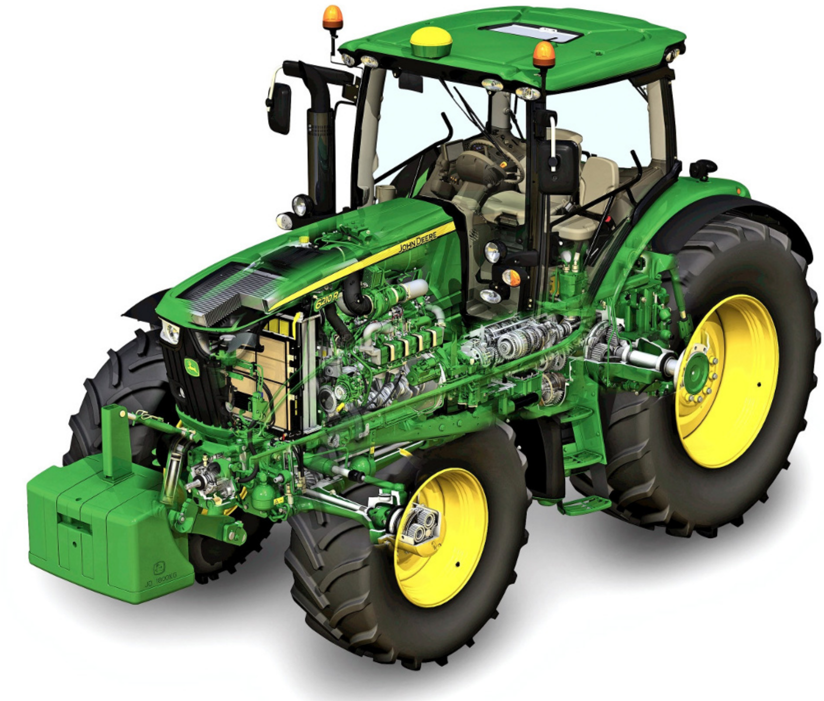


Fig.1.37: John Deere 6R lineup, 4- and 6-cylinder turbo charged diesel engines 4.5 and 6.8 l, four transmission options (here with an 8-speed dual clutch partial power shift). Rated eng. power 77–184 kW @ 2,100/min (97/68 EG, without boost) Courtesy John Deere 2017

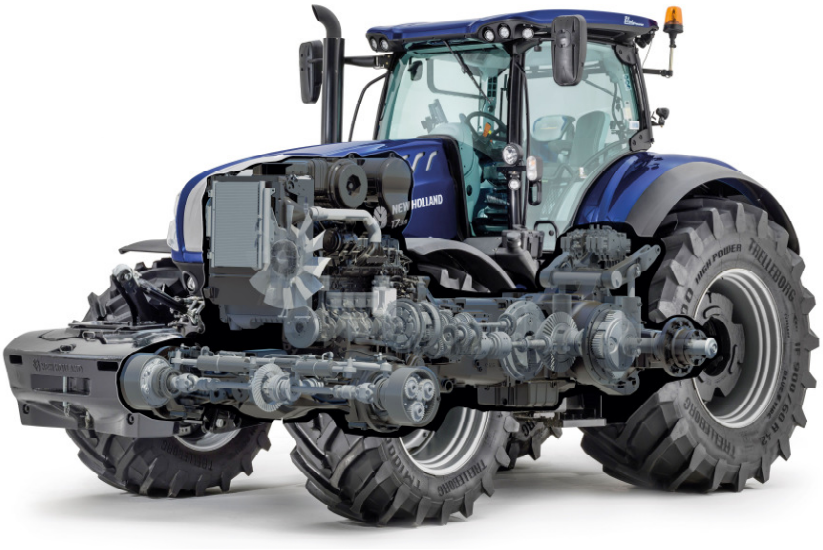


Fig. 1.38: New Holland T7 lineup with 6-cylinder 6.73 l turbo charged diesel engines. Full power shift 19/6 or CVT, rated eng. power 92–221 kW @ 2,200/min (ECE R 120). Courtesy NH 2017

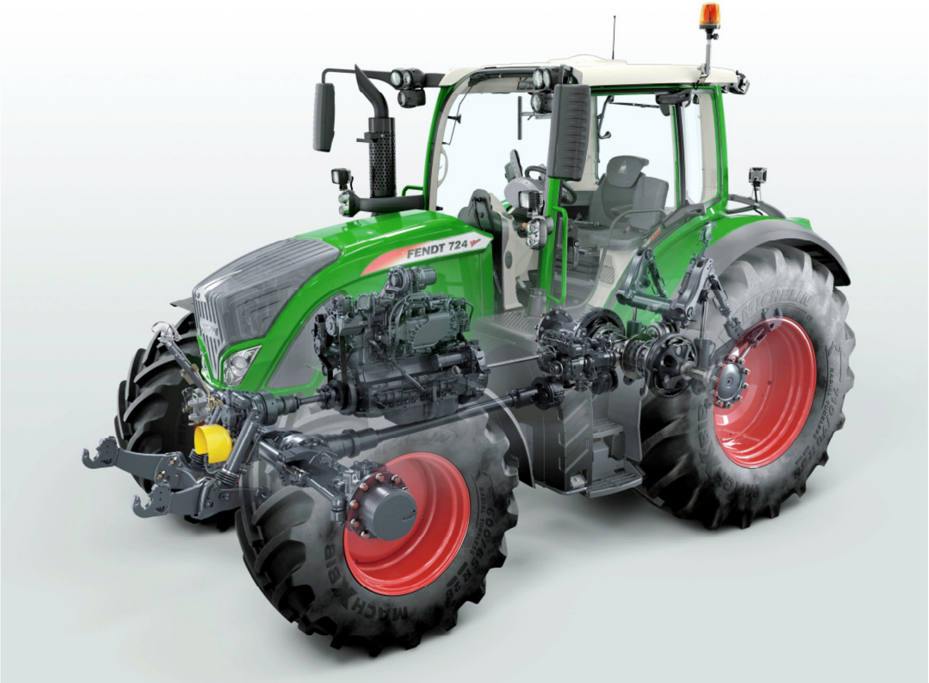
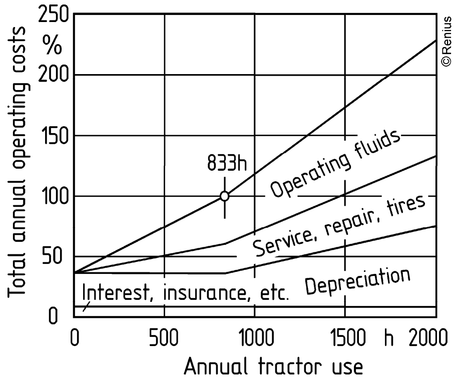


Fig. 1.39: Fendt Vario 700 lineup with 6-cylinder 6.06 l turbo charged diesel engines. CVT “Vario” with two synchronized ranges (28 and 50 km/h) and reversing by the hydrostatic Vario system. Rated engine power 106–174 kW @ 2,100/min (97/68 EG). Courtesy AGCO-Fendt 2016

1.5 Costs of tractor operation



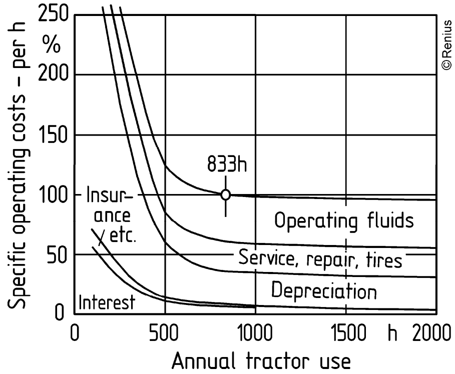


Fig. 1.40: Annual operating costs for a 100 kW tractor for technology level IV. Costs of service +repair+tires decrease for higher power but increase for lower power. Data from KTBL

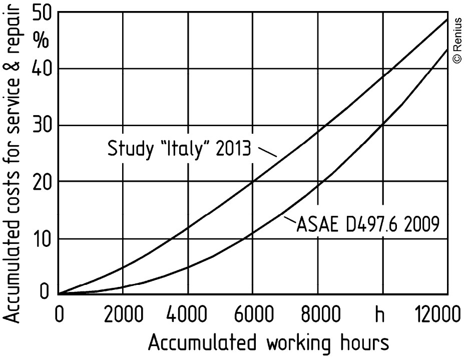


Fig. 1.41: Accumulated costs for service and repair including tires in % of tractor list price. Study of 100 4WD tractors in Northern Italy 2013 [1.33]

Operational costs of tractors are usually related to one hour of use (p. h.) and they are mainly influenced by the following factors:

- Power level
- Human influences
- Annual use in hours
- Load levels in operation
- Technology level (chapter 1.6)
- Labour costs as per working hour
- Costs for operating fluids and spare parts

Power level. The bandwidth of rated engine power has become very high, for standard tractors now from about 10 to 350 kW. The working hours for service and repair increased however less than the tractor power. Although the relative tire replacement costs increase with the power level, the relative costs for service, repair and tires in total decrease slightly with the rated power for constant labour costs p. h.

Technology level. It influences the operational costs mainly due to the different technical complexity and thus depreciation of initial tractor costs.

Annual use. The more hours per year, the higher the total annual operating costs, as shown in the upper part of **Fig. 1.40** [1.32].

However, the costs per hour decrease with increasing hours p. a., mainly due to the declining influence of the initial costs see depreciation in the lower part of Fig. 1.40.

Tractors which are overdesigned in power and specifications, create high costs.

The fraction of operating fluid costs is connected to crude oil prices and subsidies. Costs for service and repair have declined in the past [1.32], confirmed by an Italian study, **Fig. 1.41** [1.33]. Values for other countries can be higher, as, for example, identified for Tanzania [1.34].

Human influences. Service and repair costs are lower for tractors in the hands of contractors than in the hands of farmers as reported in [1.33].

1.6 Tractor markets, global product planning, and costs

Tractor markets and companies. The annual global tractor production is estimated to be almost 2 million units over about 15 engine kW (average 2012–2015). A very high number of small tractors is produced in China and India.

The countries with the largest numbers of produced tractors (in units) are:

1. India
2. Japan
3. USA
4. China
5. EU
6. Brazil
7. Turkey
8. Belarus

The value of global tractor production was about 45 bn US $ p.a. in 2017.

After World War II the number of tractor-producing companies was very high, but global competition forced a concentration to companies with higher sales.

A typical example is CNH Industrial, which is a step-by-step merger of five companies in the following sequence:

Case + IH, Ford + NH, Ford NH + Fiat = NH, Case IH + Steyr, Case IH + Steyr + NH = CNH Industrial [1.35].

AGCO was formed in 1990 by a management buyout of Deutz-Allis initiated by *Robert Ratliff* [1.8]. Current AGCO tractor brands (as of 2019) are:

MF, Fendt, Valtra, and Challenger.

Klöckner-Humboldt-Deutz (KHD) was a large German tractor producer for more than half a century but sold its tractor business to the Italian SAME Group in 1995.

CLAAS started in 1997 with the Xerion, and acquired the majority of Renault Agriculture in 2003, developing a complete tractor line with increasing market share.

John Deere remained independent and is globally leading in tractor sales (2018).

Specification by technology levels. The trend towards an internationally organized tractor business has to consider a continuously increasing bandwidth of specifications from "very simple" (mostly low power) to "very sophisticated" (mostly high power).

A typical tractor for USA or Western Europe is not suitable for Africa or India as it is over-specified and too expensive for these countries. This was the motivation for developing a structure of globally demanded tractor specifications, **Table 1.4** [1.36].

Table 1.4: Survey of worldwide demanded tractor specifications, classified by five technology levels for two-axle tractors (*Renius* 2002 [1.36], updated)

Technology level	Nominal engine power			Wheel drive			Diesel engine					Drive transmission					PTO			Hydraulics				Cab & ROPS			Elec-tronics			Level of §		
	Low	Medium	High	Only 2-wheel drive	Front drive optional	4-wheel drive std.	1 cyl.	2 cyl.	3 cyl.	4 cyl.	6 cyl.	Very simple	Simple	Partial power shift	Full power shift	Infinitely variable	540/min	540 & 1000/min	3 or 4 speeds	Rear 3-point hitch	Remote control	Rear and front hitch	Load sensing hydr.	No cab, no ROPS	ROPS / low cost cab	Comfort cab	Not existing	Low cost concepts	High tech concepts	Low	Medium	High
I	X	(X)		X			X	X	X	X		X					X			X				X			X			X		
II	X	X			X			X	X	X	X		X					X		X	X			X	(X)		X	(X)		X	(X)	
III		X	(X)		(X)	X		(X)	X	X	X			X				(X)	X	X	X	X			X	X		X	(X)		X	(X)
IV		X	X			X			X	X	X			X	X				X	X	X	X	X			X			X			X
V		X	X			X			X	X	X					X			X	X	X	X	X			X			X			X

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Table 1.5: Cost savings by tractor families, example for standard tractors of USA and Western Europe (technology levels III, IV, and V). A global structure needs additional families for low levels.

Tractor family	1	2	3	4
Power, kW (EG 97/68)	45 – 75	75 – 120	120 – 170	170 – 300
Rated eng. speed, rpm	2,000 – 2,300			
Typical wheel base, mm	2,300	2,500	2,750	3,000
Diesel engine	3 cyl.	4 cyl.	6 cyl.	6 cyl. large
Volume of functions	moderate	very high		high
Comfort level	moderate	high	very high	

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***Tractor families** are needed, **Table 1.5**, to save costs for development, tooling, logistics, and spare parts through many identical parts within each family* [1.37-1.40].

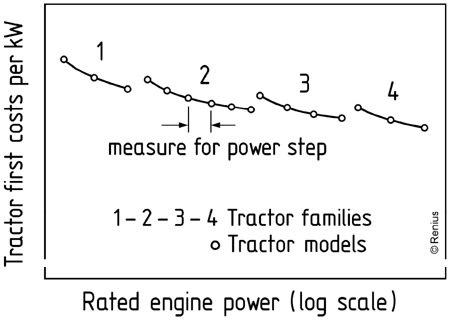

Fig. 1.42: Specific tractor production costs, typical relation for 4 tractor families. Family 2 has a very narrow power gradation in this case to cover the most popular power range

Table 1.6: Project foundation matrix. The black points indicate “action required” [1.40]

	R&D	Marketing	Factory	Finance
Product				
- Specifications	●	●		
- Production costs	●		●	●
Markets				
- Sales regions		●		
- Volume forecast		●		
- Potential revenues		●		
Investments				
- Development	●			●
- Factory			●	●
- Sales & Service		●		●
Capacity				
- R & D	●			
- Factory			●	
- Finance				●
Economic Evaluation				●

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Hence, **specific production costs** for a given family structure are not unique for all family members, **Fig. 1.42** [1.37, 1.38].

The top model of a family has the lowest specific costs (costs per kW) as it usually offers the best performance utilisation of its parts or components [1.37, 1.38]. The slope of the curves can be kept low by adjusting components to the power (axles, tires, …).

The quality of a family concept can be controlled by dividing the total number of parts by the number of models [1.36, 1.40].

Typical cost distribution by components for a 150 kW tractor with driven front axle, technology levels IV and V (Europe):

- Transmission with rear axle ....26 %
- Diesel engine with exhaust treatment 19 %
- Hydraulics and power steering ....11 %
- Cabin with integrated ROPS ....10 %
- Driven front axle (12 %), tires (7 %), electrics, frame, sheet metal, tank, etc. ... 28 %
- Assembly and quality check .... 6 %

Planning a new tractor family needs a systematic procedure, **Table 1.6** [1.41, 1.42].

If only one of the four departments is not convinced of the project, it should not be settled and better discussed again.

It has been proven to put “project managers” in charge to take care of all activities during development. The various subjects of project progress control [1.43] can be summarized in three central duties:

1. To control the status of *specifications*
2. To control the status of *all costs*
3. To control the status of the *time scale*

1.7 Seeking the best for tractor user and tractor producer

The view of the customer.

Customer needs must be the basis for all product definitions, as customers are interested in an economic investment.

A farmer considering buying a tractor usually applies the following criteria:

- Required functions: profile of operation
- Reliability and dealer service quality
- Performance, published test results
- Image of the tractor producer
- Comfort and safety
- Fuel economy
- Appearance (styling, colour, etc.)
- Resale value

Tractor functions, performance, and fuel economy depend on three groups of technical factors, **Table 1.7**.

The view of the tractor producer.

The company engaged in developing and selling agricultural tractors is interested in both: *meeting customer needs as well as possible, and making an adequate profit of the producing company.*

Therefore, the tractor producer has to make sure that its total costs are below the achievable ex factory revenues.

To know this level, he must analyze all important market factors, **Table 1.8**, left hand, and to balance them with all factors affecting its total production costs – that means ex factory costs, right hand.

Table 1.7: Factors affecting the value of agricultural tractors (upper technology levels)

Tractor operation (field and road): performance, functions, and fuel economy		
Component properties	External mechanics	Process control
- Diesel engine - Transmission - Axles, 4WD, tires - PTO concept - Hydraulics	- Tractor weight - Implement & trailer forces - Ballast weights - Axle load distribution - Terrain characteristics	- Driver information - Internal bus - External bus - Assistance systems - Automation

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Table 1.8: Tractor retail price and tractor ex factory costs are the basis for the economy of the tractor producer

Tractor *retail price* depends on	Tractor *ex factory costs* depend on
- Benefits expected by the customer	- Product volume, units (per year)
- Image of the brand	- Component flexibility within the family
- General market price level	- Grade of standardized parts
- Market balance: supply↔demand	- Design & value analysis maturity
- Quality of dealer network	- Production costs of the factory
- Quality of sales methods	- Costs for production start-up
- Target of market share	- Production life cycle (years)
- Size of dealer network	- Costs for R&D, warranty, etc.
- Emotions, styling, stories	- Costs for sales & service support
- Dealer costs and dealer margin	- Other overhead

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1.8 Standards and regulations

Standards simplify not only tractor-implement combinations but also international business. Early historical issues addressed specifications such as PTO (shaft & rpm), three point hitch and track width.

Standards were initially developed nationally for tractors (for example by ASAE in the US, BSI in the UK, or DIN in Germany). Their subjects were expanded later on to other areas such as human relations, safety, emissions, communication systems, sustainability, recycling, and others.

Regarding history, the United States has been leading in tractor standards for a long time; see, for example, the first ASAE standard for the PTO, in force in 1927 [1.23].

Six recent phases of standardization have been identified in [1.44]:

- 1950s: reduction of design variety
- 1960s: interchange ability & usability
- 1970s: ergonomics & operator safety
- 1980s: tractor & implement interfaces
- 1990s: safety (CEN) & electronics (ISO)
- 2000s: systems and processes

While technical aspects and customer demands drove the first phases, standard methods for official regulations came up in addition – with European EN standards mainly for safety and traffic regulations.
Safety standards are often published by both, EN and ISO, ensuring the acceptance on the European and international levels; see, for example, the general European safety standard EN ISO 4254-1 [1.45].

The *International Organization for Standardization* (ISO) started for tractors and agricultural machinery in 1952 [1.45]. ISO has become the most important organization of this kind because of globalization.

ISO standards have been developed for all areas of tractor design. The number of national standard projects has declined considerably; however, they are sometimes derived from ISO standards; for example, the German DIN ISO 1219-1 for hydraulic symbols or DIN EN ISO 4254-1 for safety requirements.

ISO activities are well organized by Technical Committees (TC), Sub Committees (SC), and Working Groups (WG).

International standards for agricultural tractors are administered by ISO TC 23 [1.46], mainly in SC4. The development of ISO standards takes place in steps. The following final versions should already be considered for tractor developments:

- ISO/DIS …. Draft International Standard
- ISO/FDIS … Final Draft Intern. Standard
- ISO ……………... International Standard

The creation of valuable ISO standards is hard global democratic work. It requires well educated, motivated engineers with diplomatic skills and good English knowledge in order to achieve economic compromises, also in the case of factual conflicts. The earlier a new standard is discussed, the better the chance for progress and global benefits.

If a standard does not get the necessary ISO agreement, it can become an "ISO Technical Report" (ISO/TR). A "Technical Specification" (ISO/TS) is also possible. It requires, however, consensus within an ISO Committee.

Every ISO Technical Committee and every ISO Sub Committee is represented by a secretariat. Its location can vary.

National representatives can make proposals which are usually discussed in a first phase in a working group. ISO standards are subjected to automatic revision.

ISO standards and regulations for tractors (including safety and emissions) are widely considered along with the following chapters as they influence an updated tractor design more than ever before.

Pages 276–277 of this book contain a list of ISO standards for tractors.

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2 Tractor mechanics

2.1 Speed and slope limits of practical tractor use

Off-road tractor speeds. Off-road working speeds have been increased from about 3 km/h (for horses) [2.1] to often between about 4 and 12 km/h for tractors; in the case of a very even surface for special operations (such as forage cutting) up to 15–20 km/h.

The following implements and nominal speeds are used for heavy tillage:

- Rotary power tiller (PTO)	3–5 km/h
- Subsoiler	4–6 km/h
- Plow, small tractor	4–6 km/h
- Plow, large tractor	7–9 km/h
- Cultivator, large tractor	7–12 km/h
- Disc harrow, large tractor	8–15 km/h

Speeds below 2 km/h are seldom used, often covered by special *creeper speeds*.

Speed limits for field work. Productivity can be increased by higher speeds but there are some typical limits:

- *Quality* of process and/or work output
- *Specific energy* consumption
- *Riding comfort*

Examples:

1. Seeding at increased speeds may increase the variance of seedling positions.

2. A higher working speed mostly needs more fuel per ha due to higher energy consumption for mass accelerations (e.g. soil).

3. Vibration amplitudes for a given surface increase more than linearly as speed increases. For better riding comfort, adequate suspensions, mostly soft suspended front axles and cabs, can compensate for this.

Speed limits on the road. Top speeds on the road are often about 28–32 km/h worldwide for standard tractors. EU regulations in Western Europe favour 40 km/h; however 50 km/h is popular for larger tractors – even 60 km/h as an option.

Slope limits on the road. This is usually no problem for a single tractor as its critical slope for overturning is so high that the driver will always have a reserve.

However, when the tractor is working with a front end loader, the critical slope of overturning can be much lower.

Slope limits off-road. Maximum slopes for plant production (facing erosion and machinery application) are as follows:

- Sugar beets, corn	12 %
- Potatoes	20 %
- Grain	25 %
- Forage	30 %
- Meadows	45 %

A standard tractor should be able to handle slopes of up to 36.4 % (20 °). Regarding overturning limits, this is also a typical value for the dynamic limit in the field due to obstacles or turning effects [2.2], while the static limit is typically double that value.

This results in the design rule that *all oil levels of the powertrain should be able to handle slopes of 20 ° (36.4 %) in all directions*, for the PTO more for stationary operations such as driving irrigation pumps.

Special tractor versions with a low center of gravity, wide track width, low tire diameter, and duals are used for strong slopes. *In this case, oil levels should be able to handle about 30° in all slope directions.*

Traction limits on slopes can be moved up by a special chassis design, **Fig. 2.1**.

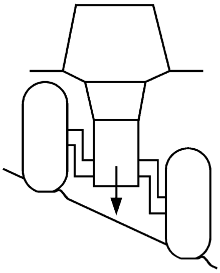

Fig. 2.1: Special tractor with slope compensation by special axles, proposed in [2.3]. No series production for tractors but realized for combines

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K. Th. Renius, *Fundamentals of Tractor Design*,
https://doi.org/10.1007/978-3-030-32804-7_2

2.2 Tire-soil mechanics

2.2.1 Basic soil properties

Importance. The energy losses between traction tires and the soil in field operations usually represent the highest share of all energy losses between the diesel engine and a pulled implement. Many international publications have been presented, often by members of the *International Society for Terrain-Vehicle Systems (ISTVS),* founded by *Mieczyslaw Gregory Bekker* (1905–89) [2.4, 2.5, 2.6], who even contributed to the design of the Apollo 15 Lunar Rover.

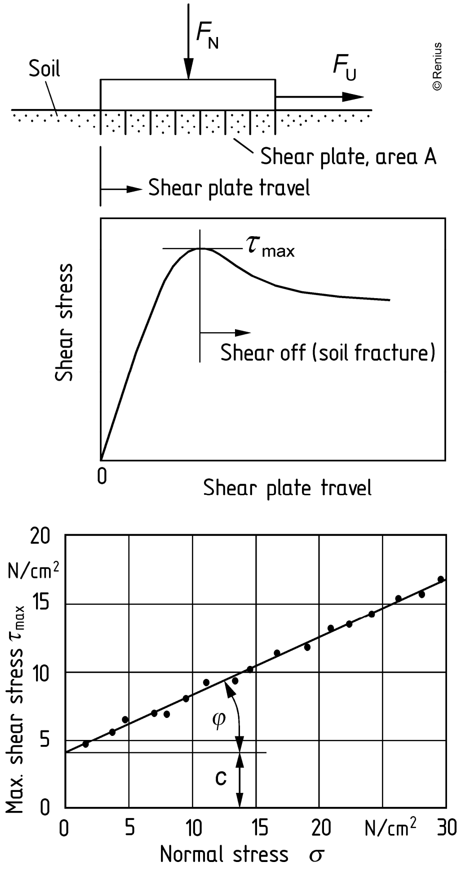

Fig. 2.2: Experiment to determine the coefficient of cohesion *c* and the angle of inner friction φ. Example for a soil with high clay content

Initial basis: the Coulomb equation. *Charles A. Coulomb* (1736–1806) published an equation modelling the friction in the case of cohesion [2.7].

This famous equation can be derived from a simple experiment, **Fig. 2.2**. The soil is loaded by a shear plate in order to measure the horizontal shear force under a defined vertical load [2.5, 2.6, 2.8, 2.9].

The shear plate is moved and the shear force F_T measured. The force values are divided by the plate area A receiving shear stress τ. This is plotted versus the plate travel for different vertical loads F_N, mid diagram. The maximum shear stress values τ_{max} are introduced into the lowest diagram of Fig. 2.2, plotting all maxima τ_{max} versus the normal stress σ.

A straight line can usually be drawn through the test results representing the equation of *Coulomb*:

$$\tau_{max} = c + \sigma \cdot \tan\varphi = c + \sigma \cdot \mu \tag{2.1}$$

In the case of no stress σ (no load), the shear stress is only carried by the cohesion c of the soil. The additional shear stress depends on the normal load, its increase with σ gives the angle of internal soil friction φ.

The term tan φ is also known as the classic coefficient of friction μ.

In terms of forces:

$$F_{U,\,max} = c \cdot A + F_N \cdot \tan\varphi \tag{2.2}$$

Cohesive soils: $c \approx 2\ldots4$ N/cm² [2.10].

Example: maximum gross traction of a tractor tire on sandy loam (estimated):

Adhesion tire-soil $c =$ 2.5 N/cm² = 25 kPa
Contact area under load = 3,000 cm²
Tire load 30 kN = 30,000 N
Angle of internal friction $\varphi = 24°$

$F_{U,max} = 2.5 \cdot 3{,}000 + 30{,}000 \cdot 0.445 = \mathbf{20.9}$ kN

Several models of shear stress τ versus plate travel have been developed [2.11].

2.2.2 Tire-soil interaction

Soil movement under traction tires has been investigated by *Walter Söhne* (1913–2011) [2.12], **Fig. 2.3**. He inserted vertical layers of white gypsum into vertical soil cuts and studied their movement under loaded tractor tires for several cases.

The loaded tire affects a certain vertical rut depth and additional horizontal plastic soil deformation.

A pulled tire without braking forces creates a certain horizontal "negative" deformation, but has no shear-off area.

When a pull force is applied and increased, local shear stress arrives at its maximum (see equation 2.1), beginning at the contact run out. The higher the traction force and the slip, the larger the area of soil shear-off.

The development of shear areas has also been studied by *Söhne* from a vertical view, **Fig. 2.4**. The higher the slip, the larger the measured shaded shear area. When the whole contact area is sheared off, the tire achieves its maximum pull; for example, in the case of cohesive soils for slip values often around or above 50 %.

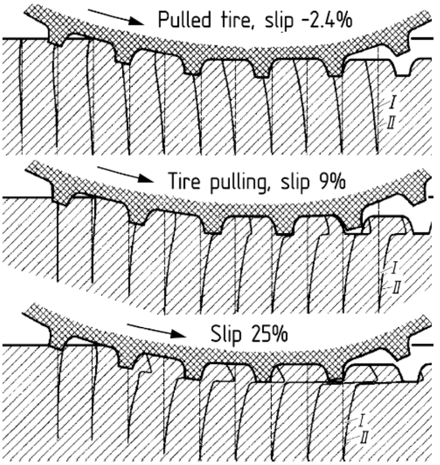

Fig. 2.3: Soil movement under tractor tires. The gypsum layer I indicates undisturbed soil which is deformed and partially sheared off by horizontal forces. Tire size 9.5-40 [2.12]

For this case, the maximum gross traction (equation 2.2) can be used to derive the maximum net traction F_{Tmax}:

$$F_{\text{Tmax}} = \mathrm{c} \cdot A + F_{\text{N}} \cdot \tan\varphi - F_{\text{R}} \qquad (2.3)$$

where

F_{N} = tire load

F_{R} = tire rolling resistance.

Maximum net traction forces are important for short-term "emergency traction", but not for permanent operation.

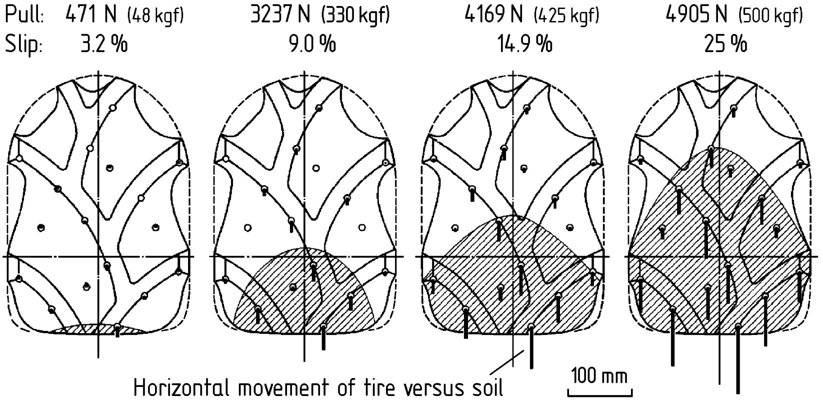

Fig. 2.4: Soil movement within the contact area of a pulling tire 12.4-24 for increased traction (from left to right). The higher the traction, the larger the area of sheared soil (shaded) [2.12]

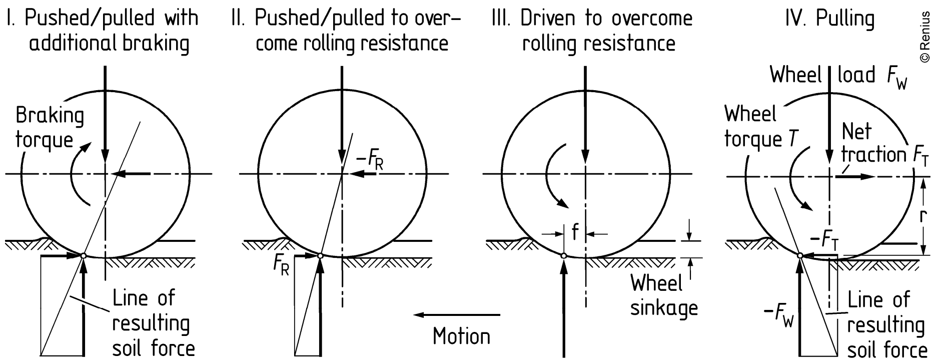
I. Pushed/pulled with additional braking
II. Pushed/pulled to overcome rolling resistance
III. Driven to overcome rolling resistance
IV. Pulling
© Renius
Braking torque
Line of resulting soil force
$-F_R$
F_R
Motion
f
Wheel sinkage
Wheel load F_W
Wheel torque T
Net traction F_T
r
$-F_T$
$-F_W$
Line of resulting soil force

Fig. 2.5: The four characteristic cases of tire-soil mechanics off-road, flat terrain [2.13]

Soil-tire mechanics can be classified by four cases [2.13, 2.14], **Fig. 2.5.**

The *two most important* ones are:

***Case II*:** a non-driven front wheel of a 2WD tractor or a wheel of a trailer.

Case IV: a pulling wheel.

A (fictive) "gross traction" F_U can be derived by **Fig. 2.6** [2.13, 2.14].

$$F_U = F_T + F_R = \frac{T}{r} \tag{2.4}$$

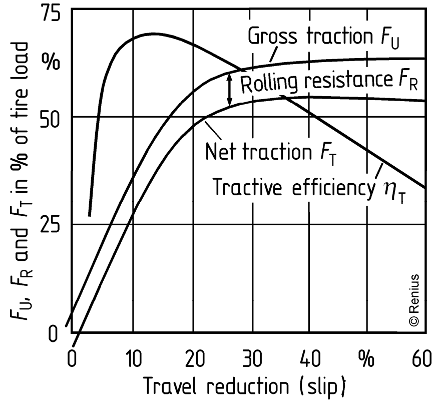

Fig. 2.6: Gross traction, net traction and traction efficiency (dry uncultivated stubble)

The radius r of F_T (Fig. 2.5) is not exactly identical with the rolling radius [2.15].

Definition of travel reduction (slip):

$$\text{Slip } i = \frac{\text{travel without slip} - \text{true travel}}{\text{travel without slip}} \tag{2.5}$$

Definition of "zero slip" by Fig. 2.5.

A simple, clear physical "law" is not possible [2.16]. Four agreements are common:

1. Zero slip *at zero net traction, case III.* Very easy to handle but determination only useful for very firm soils (a wheel can spin without traction under muddy conditions!).
2. Zero slip *between II and III.* Several authors recommend measuring the travel distance for a given number of revolutions for II and III and calculating the effective tire radius from the mean distance [2.14].
3. Zero slip *on concrete at zero pull* with and without correction factors [2.17].
4. Effective radius *derived from tire data.*

Traction and rolling resistance modelling can be simplified by coefficients such as κ (net traction) and ρ (rolling resistance):

$$\kappa = \frac{\text{Net traction } F_T}{\text{Vertical tire load } F_W} \tag{2.6}$$

$$\rho = \frac{\text{Rolling resistance } F_R}{\text{Vertical tire load } F_W} \tag{2.7}$$

Calculation of driving torque T:

$$T = (F_T + F_R) \cdot r = (\kappa + \rho) \cdot F_W \cdot r \tag{2.8}$$

Tractive efficiency:

$$\eta_T = \underbrace{\frac{\kappa}{\kappa+\rho}}_{\eta_\rho} \underbrace{(1-i)}_{\eta_i} = \eta_\rho \cdot \eta_i \tag{2.9}$$

The **four characteristic cases** of tire-soil mechanics (Fig. 2.5) result in performance curves as plotted in **Fig. 2.7** [2.14].

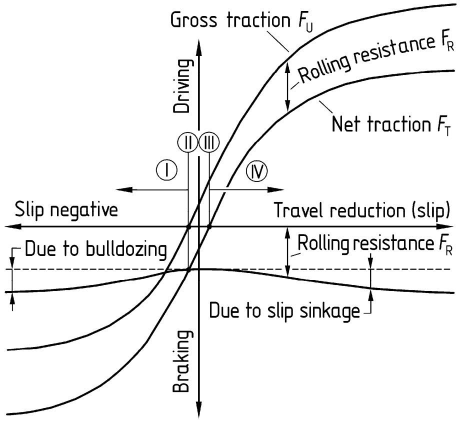

Fig. 2.7: Traction, rolling resistance and braking forces versus travel reduction after *Söhne* [2.14], Sections I to IV as defined by Fig. 2.5

In the case of braking (left part), the rolling resistance supports the retardation and the high negative net traction is a measure of the tire braking force.

Fragmentation of travel reduction and rolling resistance is shown in **Fig. 2.8**.

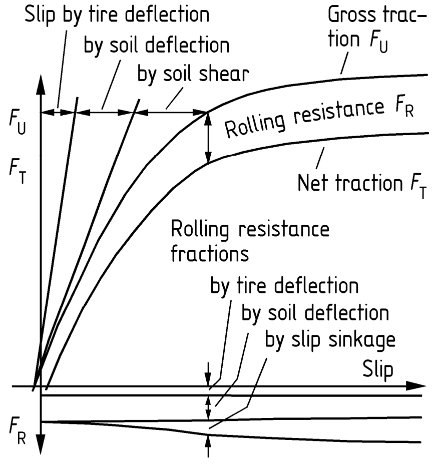

Fig. 2.8: Traction forces, travel reduction (slip), and rolling resistance, after *Söhne* [2.14]

Soil-tire interaction during turns is characterized by lateral forces (side forces).

Fig. 2.9 shows the soil force and its components F_R and F_L of a non-driven tire under a certain slip angle for off-road conditions (tractor front axle).

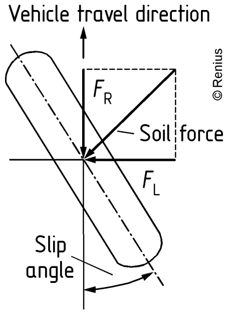

Fig. 2.9: Horizontal soil force and its components side force F_L and rolling resistance F_R of a steered, non-driven, loaded tire on soft soil

If the steering wheel is driven, the soil force vector can be estimated by $\kappa \cdot F_W$. The traction force is reduced as side forces increase [2.18 - 2.20]. This is well known for road surfaces ("Kamm's Circle") but was also demonstrated for off-road conditions, **Fig. 2.10**. It is interesting that the side forces have their maximum not at zero forward slip but at a negative value of 5–10 % (braking mode).

Soil conditions have a higher influence than the size of the traction tires [2.19].

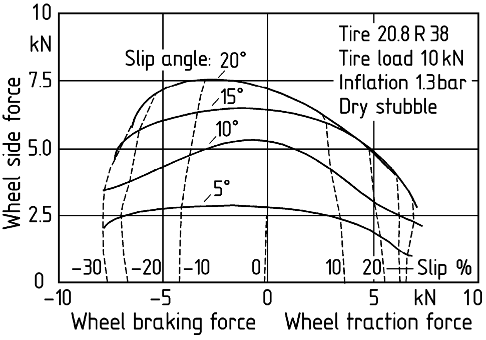

Fig. 2.10: Off-road wheel forces under side slip. Measurements with a large radial tire [2.19]

Equation (2.9) indicates that tractive efficiency can be understood as a product of

- η_p ...*rolling resistance efficiency* and
- η_i ...*travel reduction (slip) efficiency.*

The author sees no clear priority as to which energy loss must be calculated first.

The zero-slip problem can be avoided by plotting tractive efficiency versus net traction [2.21] or its coefficient [2.22], **Fig. 2.11**.

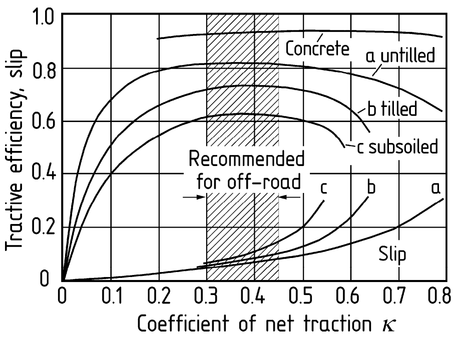

Fig. 2.11: Tractive efficiencies, 20.8R42, duals, 83 kPa tire inflation pressure, dry soil [2.22]

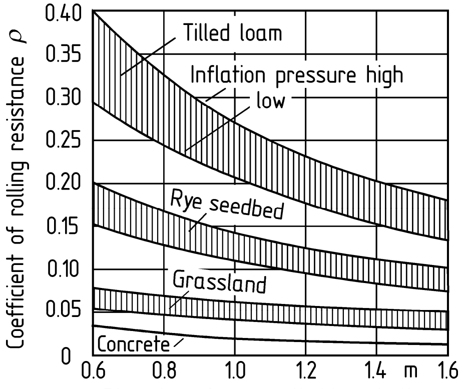

Fig. 2.12: Rolling resistance 1940 [2.23]

Large tires favour low losses, **Fig. 2.12** [2.23].

Values ρ of traction tires on concrete are 2-3% (bias) and 1.5-2.5% (radials); influenced by *dimensions, load, inflation* [2.24]. Net traction coefficients κ were presented by *Söhne* [2.21] for bias tires. They are updated by the author for radials in **Fig. 2.13**.

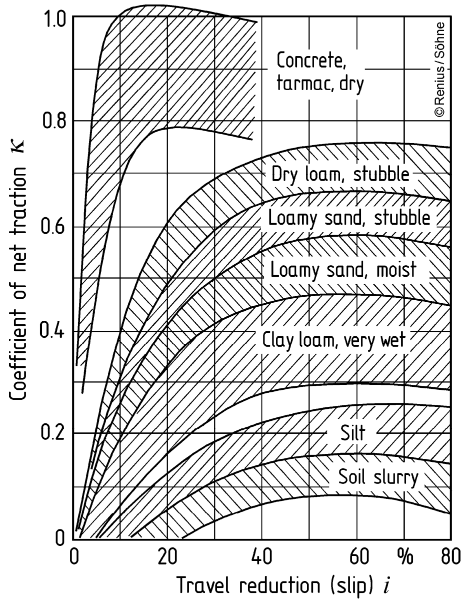

Fig. 2.13: Net traction coefficients for radials, estimated by the author (values for bias lower [2.21])

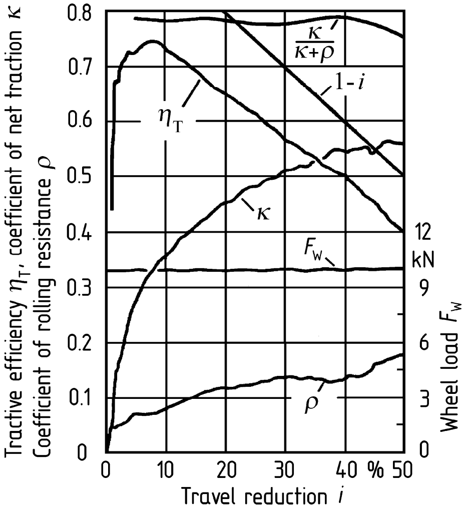

Fig. 2.14: Example for analog real time measurement of traction performance. Bias ply tire 16.9-30 on dry stubble, 1.4 bar [2.25]

Traction mechanics were measured in real time by *Steinkampf*, **Fig. 2.14** [2.25].

2.2.3 Traction models

Traction models for bias tires. A simple model uses for dry stubble only the slip i as input for the net traction coefficient κ:

$$\kappa = \sqrt{i} \tag{2.10}$$

An updated review on traction mechanics was presented in [2.26]. The typical basis of semi-empirical equations is often an e-function based on the Coulomb equation. *Janosi* and *Hanamoto* [2.27] simplified former models of *Bekker*:

$$\tau = (c + \sigma \tan \varphi) \cdot (1 - \mathrm{e}^{-\frac{j}{\mathrm{K}}}) \tag{2.11}$$

where

τ, c, σ and φ from equation 2.1 and Fig. 2.2
e = natural base = Euler constant = 2.71828…
j = shear displacement
K = factor addressing initial slope of $\tau(j)$

This often-cited equation was originally presented for tracked vehicles, but also became a base for pneumatic tires, leading to similar equations with travel reduction i. It is only valid up to slip values near $\tau_{\max}$.

Dwyer demonstrated that the basic form of (2.11) is even useful for traction predictions on concrete (for bias tires) [2.28]:

$$\kappa = 1 - \mathrm{e}^{-20i} \tag{2.12}$$

with κ and i as already defined.
Example: slip $i = 0.05 \rightarrow \kappa = 0.63$

Also the popular model of *Wismer* and *Luth* [2.29] addresses bias ply rating tires:

$$\kappa = \underbrace{0.75 \cdot (1 - \mathrm{e}^{-0.3\, C_{\mathrm{N}} \cdot i})}_{\text{Gross traction } \mu} - \underbrace{(\frac{1.2}{C_{\mathrm{N}}} + 0.04)}_{\text{Rolling res. } \rho} \tag{2.13}$$

with slip i and "wheel numeric" C_{N}:

$$C_{\mathrm{N}} = \frac{b \cdot d \cdot CI}{F_{\mathrm{W}}} \tag{2.14}$$

where

b = tire width unloaded
d = tire diameter unloaded
CI = ASABE cone penetrometer index
F_{W} = tire load

Equation (2.13) was later modified by several authors, in particular by *Brixius* [2.30]. For bias ply tires, he proposed the equation (2.15):

$$\kappa = 0.88\,(1 - \mathrm{e}^{-0.1 B_{\mathrm{n}}}) \cdot (1 - \mathrm{e}^{-7.5 i}) - \left[\frac{1}{B_{\mathrm{n}}} + \frac{0.5 i}{\sqrt{B_{\mathrm{n}}}}\right]$$

in which the "mobility number" B_{n} is:

$$B_{\mathrm{n}} = \frac{b \cdot d \cdot CI}{F_{\mathrm{W}}} \cdot \frac{1 + 5\, \delta/h}{1 + 3\, b/d} \tag{2.16}$$

where

CI = ASABE cone penetrometer index
b = tire width unloaded
d = tire diameter unloaded
F_{W} = tire load
δ = vertical tire deflection under load
h = tire section height

Typical values of the cone index CI [2.30]:

- Wet soft or loose sandy soils ... 450 kN/m^2
- Stubble at springtime ……..….. 700 kN/m^2
- Dry soil at wheat harvest ……. 1200 kN/m^2
- Heavy dry soil, uncultivated .... 1800 kN/m^2

Values of B_{n} (soft–firm) are about 15–40.

Separate modelling of the rolling resistance coefficient ρ is presented by the equation

$$\rho = \frac{1}{B_{\mathrm{n}}} + \frac{0.5 i}{\sqrt{B_{\mathrm{n}}}} + 0.04 \tag{2.17}$$

Traction models for radials. Radials widely replaced bias tires at the upper technology levels. *Zoz* and *Grisso* [2.22] picked up statements of Brixius [2.30] recommending the following modified *Brixius* equation for radials (equ. 2.15a):

$$\kappa = 0.88\,(1 - \mathrm{e}^{-0.08 B_{\mathrm{n}}}) \cdot (1 - \mathrm{e}^{-7.0 i}) - \left[\frac{1}{B_{\mathrm{n}}} + \frac{0.5 i}{\sqrt{B_{\mathrm{n}}}}\right]$$

As radials have a slightly lower rolling resistance, equation (2.17) was modified to

$$\rho = \frac{1.2}{B_{\mathrm{n}}} + \frac{0.5 i}{\sqrt{B_{\mathrm{n}}}} + 0.03 \tag{2.17a}$$

The newly recommended factors within (2.15a) and (2.17a) are: 0.08 – 7.0 – 1.2 – 0.03.

A detailed traction model was also presented by *Upadhyaya et al* [2.31], the (simpler) Hohenheim model by *Schreiber* [2.32].

2.2.4 Multi-pass and inflation control

Multi-pass of tires on soil increases traction forces and traction efficiency.

Performance characteristics of tractor tires are usually presented for unaffected soils. Correction should be made in the case of multi-pass; for example, for the second axle of a 4WD tractor off-road.

The physical effect is a reduction of rolling resistance and an increase of the net traction at the same slip. Hence, wheel torque is not very sensitive to multi-pass and cannot be used to address it as both effects result in a certain compensation.

Table 2.1: General importance of tire inflation pressure addressing on-road and off-road. The crosses X indicate positive influence

Criteria	Inflation pressure high e. g. 2 bar	 low e. g. 0.6 bar
Initial cost per load unit	X	
Required vehicle space	X	
Rolling resistance road	X	
Tire weight	X	
Keeping on track	X	
Soil compaction		X
Off-road performance		X
Riding comfort, peak forces		X
Dynamic overturn risk		X

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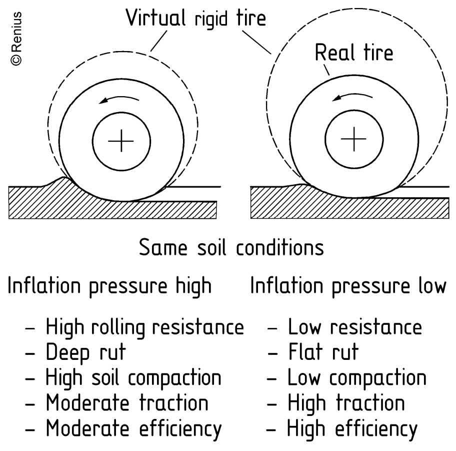


Fig. 2.15: Influence of tire inflation pressure under field conditions

The reduced rolling resistance of multi passing was recognized early, along with the introduction of pneumatic tires as, for example, expressed by *H. Meyer* in 1933.

Special research on multi-pass wheel mechanics has been carried out by *Holm* [2.33, 2.34]. He measured traction coefficients in a soil bin in a sequence of up to four passes and found the main increase during the second pass for both, the coefficient of net traction and the traction efficiency.

Regarding a 4WD tractor he derived:

$$\kappa_1 + \rho_1 \approx \kappa_2 + \rho_2 \qquad (2.18)$$

index 1 - front axle
index 2 - rear axle
with $\kappa_2 > \kappa_1$, at all times

This simplified model supposes that gross traction (and driving torque) is not influenced by multi-pass for the second pass.

Holm's multi-pass research results have been confirmed in principle by later publications, such as [2.35, 2.36].

The ratio of κ_1 to κ_2 depends on the tire dimensions and soil conditions.

The author proposed for dry uncultivated soil and same slip front and rear the following equation as a guideline [2.37]:

$$\kappa_2 \cong 1.1\,\kappa_1 \qquad (2.19)$$

Taylor et al. presented a factor of even 1.3 for tilled soil at 10% slip for a 13.6-38 [2.36].

A well-adjusted tire inflation pressure means, as a general rule, *high values on-road and low values off-road*, **Table 2.1**.

The benefits of low pressures off-road were already reported in 1935 [2.38].

Tractor tires have been developed by the related industry for an unusually large bandwidth of specified inflation pressures in order to offer an adequate adjustment to the surface and speed conditions.

This potential should be used – mainly for larger tractors. The advantages are convincing, **Fig. 2.15**.

These **advantages of tire inflation control** have been reported in many publications.

Fig. 2.16 shows results from the former soil bin of the TU Munich [2.39]. The adjusted soil conditions simulate a field structure which is similar to that of a stubble field. Increased net traction and decreased rolling resistance benefits both, pull and tractive efficiency.

The gain in traction (at constant slip) and the improved tractive efficiency were also reported in [2.22] with benefits in this case mainly for higher net traction coefficients, **Fig. 2.17**. It may be noted that a pressure of 0.41 bar is already very low. Such values generate very low soil compaction, and rut depth, **Fig. 2.18**; however, optimum pressures for net traction coefficients can be slightly higher, as shown in the following diagrams.

Wide section tires allow inflation pressures as low as 40–60 kPa (0.4–0.6 bar), which can be applied off-road even for high tire loads, as specified load capacities increase with decreased working speeds.

Field experiments have been carried out over many years at Fachhochschule Südwestfalen-Soest (Germany) under *L. Volk,* “teaching” the benefits even by public demonstrations.

Are there higher loads for the transmission through increased net traction?

This question addresses

- multi-pass and/or
- reduced tire inflation pressure

Torques are only very slightly affected, as both the reduced rolling resistance and the reduced tire radius largely compensate for higher net traction ratios [2.33, 2.39].

This is an important finding, as both effects improve productivity, but do not require reinforcements of the transmission.

This also means that off-road torque measurements at the drive axles cannot monitor net traction changes well.

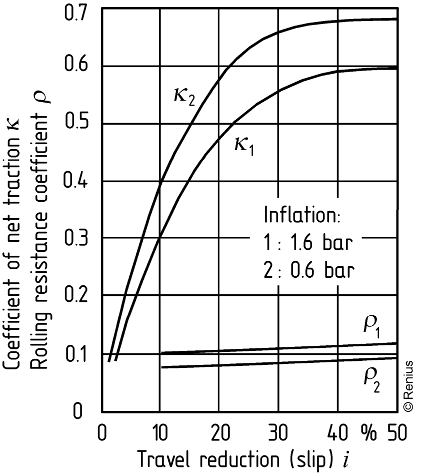


Fig. 2.16: Typical benefits of reduced tire inflation pressure. Soil bin measurements with tire 18.4R38 at a load of 26.2 kN.
Rolling radii: r_1 = 0.858m, r_2 = 0.812m [2.39]

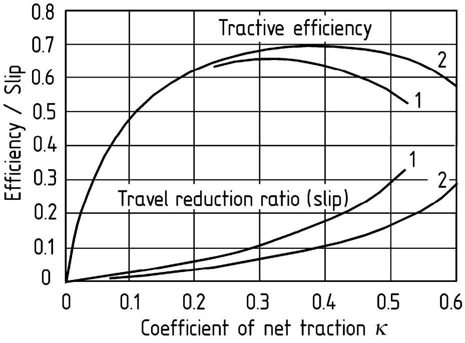


Fig. 2.17: Increased tractive efficiency by lower tire inflation pressure (2). Tire size 710/70R38, tilled loose soil.
Tire inflation: 1 – 1.93 bar, 2 – 0.41 bar [2.22]

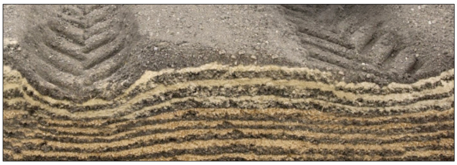


Fig. 2.18: Reduced rut depth by low section tires and low inflation pressures. Courtesy *L. Volk* in cooperation with Michelin

The following two diagrams represent measurements of *Rempfer* [2.39] - **Fig. 2.19** with results for subsoiling and two tire dimensions. The reduction of the tire inflation of the standard tire delivered the best efficiency at 80 kPa, but not below.

The reason is that the nominal load capacity was exceeded at lower pressures.

The wide section tire (below) offered a considerably higher tractive efficiency and was at the same time less sensitive to problems of very low pressures.

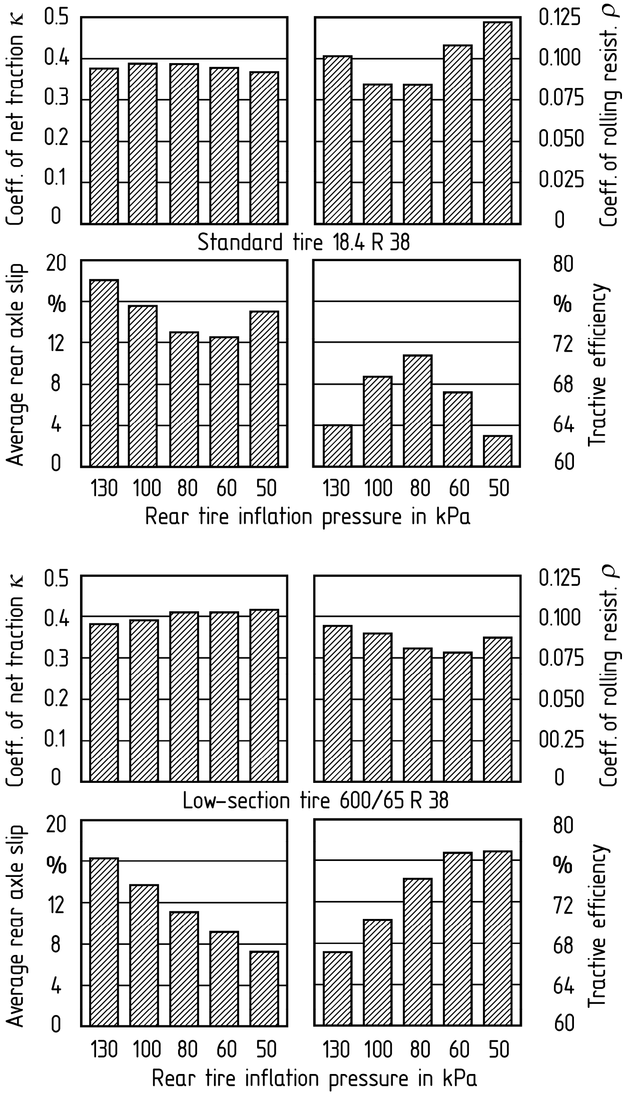


Fig. 2.19: Terramechanics of a 4WD tractor rear axle. Full load, subsoiling with 7.4 km/h. Reduced tire inflation pressures considerably improved productivity and energy efficiency [2.39]

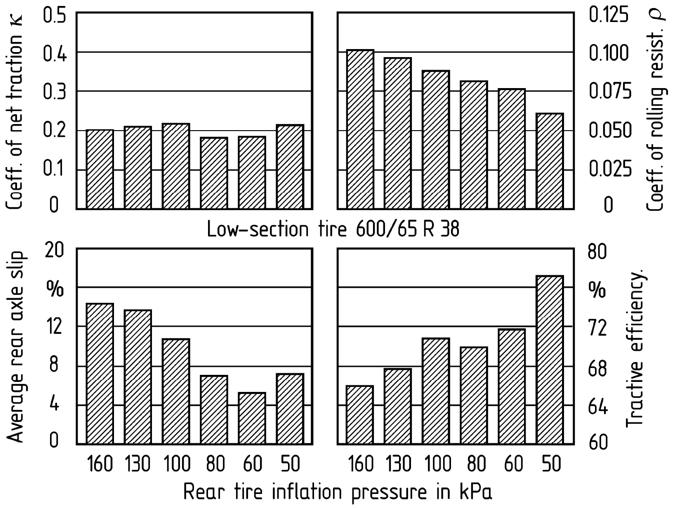


Fig. 2.20: Terramechanics of a tractor rear axle for harrowing at about 11 km/h [2.39]. Reduced tire inflation pressures considerably improved productivity and energy efficiency

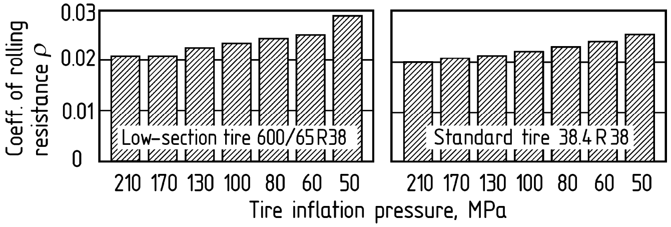


Fig. 2.21: Rolling resistance on concrete at 15 km/h: high inflation pressures are beneficial [2.39]

Harrowing at 11 km/h needs low traction ratios, **Fig. 2.20**, but also has important benefits of low inflation pressures due to reduced slip and rolling resistance.

Fig. 2.21 underscores Table 2.1: High inflation pressures reduce rolling resistance for on-road operations [2.39]. High differences result for low-section tires (left)!

Belt drives achieve higher off-road traction efficiencies than tires, mainly for high pull forces, **Fig. 2.22** [2.22]. They are, however, less suitable on the road.

The distinct element method DEM uses multi-body mechanics defining each soil particle by physical properties modelling soils and soil-tire systems [2.40, 2.41].

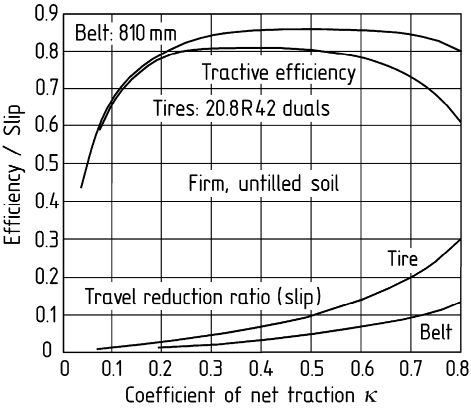


Fig. 2.22: Belt and wheel performance for typical US primary tillage operations. Evaluation by tractive efficiency and travel reduction (slip) [2.22]

2.3 Soil compaction under tires

2.3.1 Soil compaction phenomena

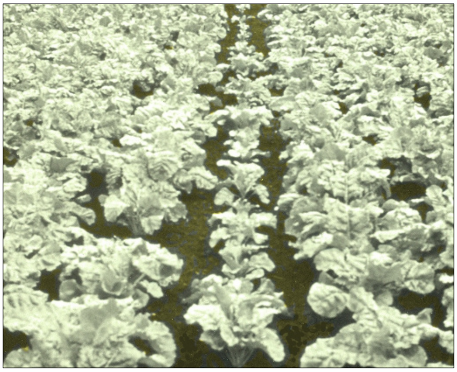


Fig. 2.23: Poor plant development (sugar beets) due to soil compaction. Photo *Golisch*

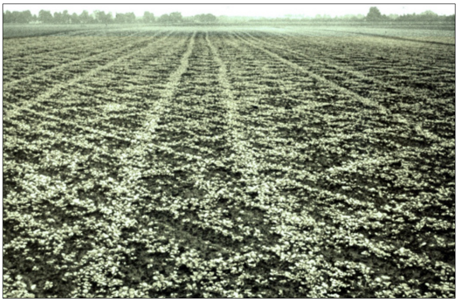


Fig. 2.24: Poor recompaction after tillage, better plant growth in the tracks. Photo *Golisch*

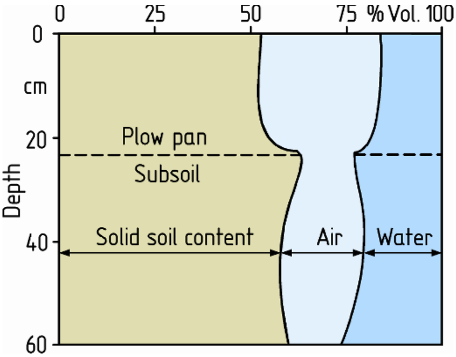


Fig. 2.25: Structure of agricultural soil, plowed in fall, investigated in spring. After *W. v. Nitsch*

Farming requirements. The physical soil structure highly influences profitable yields. If the soil is compacted too much – for example by traffic on the field under wet conditions – biological activities are poor, affecting plant growing, **Fig. 2.23**.

If the soil is too loose, water from deeper layers cannot rise due to poor capillary action, **Fig. 2.24**.

This can happen when seeding directly follows after deep tillage. In this case, recompaction is necessary after tillage using a soil compactor which also cracks larger agglomerations.

Optimum conditions can be found between these two extremes. They are addressed by the porosity n which is defined as

$$n = \frac{\text{Pore volume (gas+water)}}{\text{Total volume}} \qquad (2.20)$$

A typical profile is shown in **Fig. 2.25**. This structure is favourable within a depth of about 20 cm, but not below. The compacted layer is typical for plowing (mostly with one side of the tractor driving in the furrow). It can be abolished by subsoiling.

Typical values for *n* are as follows:

- Just plowed .... 60 %
- Plowed in fall, examined in spring .... 50 %
- Optimum for plant growing ..... 43–48 %
- Recommended: .... at least 38–40 %

The low recommended values are valid for light sandy soils while the higher ones address heavy loamy or clayey soils.

Evaluation of porosity values should be combined with the measured distribution of pore sizes [2.42].

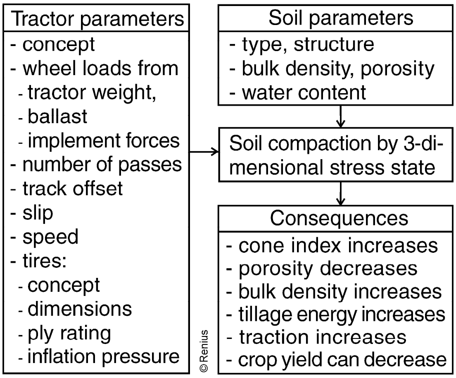
The load on tires in the field is the first item of a typical causal chain:

Load → soil stress → soil compaction

Highly compacted soils are often found in the field headland sections as evidenced by poor water drainage due to high axle loads with lifted mounted implements.

Many parameters influence soil compaction; for all its consequences see **Table 2.2**.

Table 2.2: Soil compaction



Tractor parameters

- concept
- wheel loads from
  - tractor weight,
  - ballast
  - implement forces
- number of passes
- track offset
- slip
- speed
- tires:
  - concept
  - dimensions
  - ply rating
  - inflation pressure

Soil parameters

- type, structure
- bulk density, porosity
- water content

Soil compaction by 3-dimensional stress state

Consequences

- cone index increases
- porosity decreases
- bulk density increases
- tillage energy increases
- traction increases
- crop yield can decrease

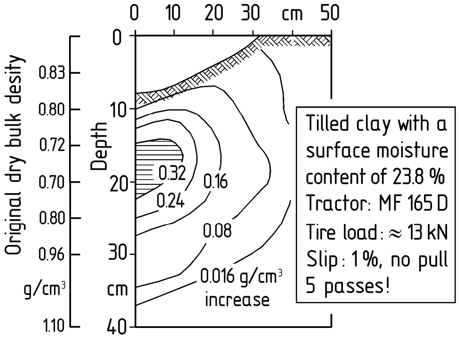


Fig. 2.26: Increase of dry soil bulk density for five passes on heavy, wet soil [2.43]

High compaction favours traction but can reduce yields. **Fig. 2.26** shows a typical dry bulk density increase and rut depth after five passes of the rear tires of a MF 165 D tractor on a tilled field [2.43].

The higher the tractor speed, the lower the compaction due to the flow resistance of the squeezed gas or water [2.44].

Söhne calculated soil stresses, enhancing fundamentals of *Fröhlich* [2.45]. **Fig. 2.27** reports the influence of water content [2.46] and **Fig. 2.28** that of the load [2.47], both for uniform inflation pressure.

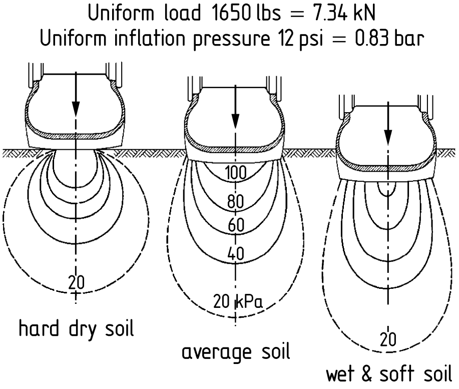

Uniform load 1650 lbs = 7.34 kN
Uniform inflation pressure 12 psi = 0.83 bar
20
hard dry soil
100
80
60
40
20 kPa
average soil
20
wet & soft soil

Fig. 2.27: *Söhne's* "pressure bulbs": Soil influence on soil stress. Calculated lines of constant principle (polar) stress under a tractor tire 12.4-28 for uniform load and inflation pressure [2.46]

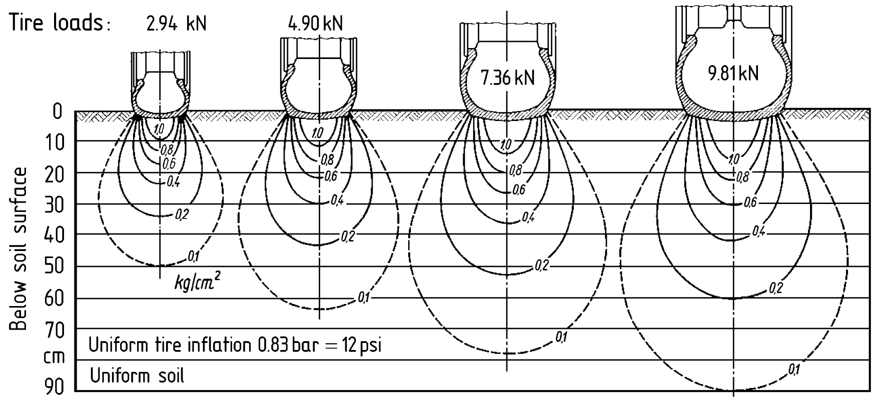


Fig. 2.28: *Söhne's* "pressure bulbs": Influence of tire load at constant inflation pressure [2.47]

2.3.2 Soil compaction control

Soil compaction under tires has been simulated by *Söhne* [2.47] with a compaction cylinder, **Fig. 2.29** and **Table 2.3**. The dominating parameters are the vertical pressure and the soil moisture. The example with a vertical pressure of 1 kp/cm^2 (14.2 psi) at 20 % water content shows that the soil III is compacted to a porosity of 36 %, which means: *critical for plant growth!*

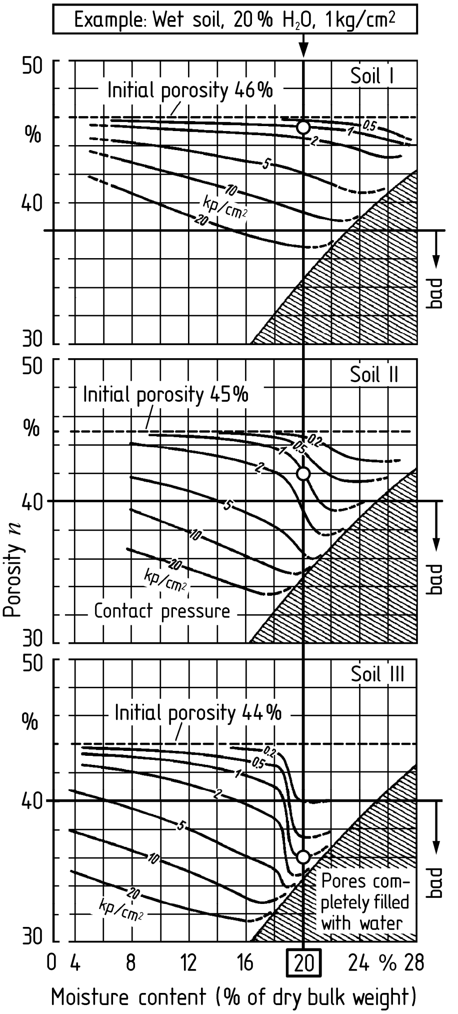


Fig. 2.29: Results of soil compaction in a cylinder with a loaded piston by *Söhne* [2.47]

Table 2.3: Grain sizes of the soils of Fig. 2.25. Particle sizes are classified in ISO 14688:2017

	Coarse sand Ø1.00-0.20mm	Fine sand 0.20-0.02mm	Silt 0.02-0.002mm	Clay under 0.002mm
Soil I	12%	54%	26%	8%
Soil II	18%	25%	31%	26%
Soil III	45%	37%	12%	6%

Soil strength measurement. Several traction models as presented in this chapter use the cone index CI describing the soil strength as derived from the vertical penetration force and projected area of a defined solid cone. The standardized ASABE soil cone penetrometer [2.48] can be said to be the most used instrument, **Fig. 2.30**.

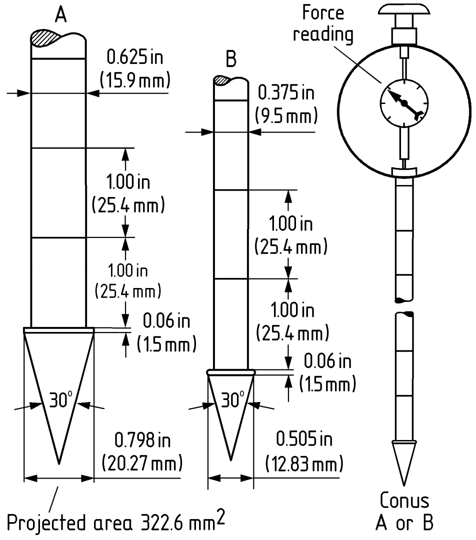


Fig. 2.30: ASABE penetrometer

Its operation was initially done by hand. A flexible steel ring and its vertical deformation was a measure for the applied penetration force. Handling of today's systems is improved by motor-driven cone penetration, electronic force measurement, and automatic CI calculation.

The soil cone penetrometer cannot be used for soils with a high content of stones.

The **penetration speed** has an influence on the penetration force, **Fig. 2.31** [2.49]. ASABE EP 542 explains practical cone penetrometer handling and recommends a penetration speed of about 3 cm/s.

Soil density measurements with the cone penetrometer? This has been discussed or tried but is not recommended in general as the soil strength and thus the cone force values are extremely sensitive to the soil moisture content, **Fig. 2.32** [2.50].

The better method of investigating compaction is to cut out soil (core) specimens and to determine the reduction of porosity.

Density can also be measured with X-rays as already in 1962 applied by *Chancellor et al.* [2.51].

Special sensors have been developed to identify soil stress under tires.

In the case of stress measurement, it is always important to define the stress direction.

Bolling developed a hydrostatic system at TU Munich, **Fig. 2.33** [2.52].

A soil bore is drilled first in order to insert the sensor without influencing the soil.

The hydrostatic system is slightly preloaded to get a perfect contact. The system is closed without remaining air. A loaded tire can pass above the balloon, the pressure is recorded. It indicates an "average" soil stress, well proved for comparisons.

Bolling has estimated absolute stress values with correction factors.

Practical recommendations to prevent harmful soil compaction under tires have been presented by a guideline of VDI-MEG [2.53] addressing soil type, moisture, and mean contact pressure, **Fig. 2.34**.

Hakanson presented another comprehensive survey [2.54] and an international team developed the free and now popular software "Terranimo" (see IoT).

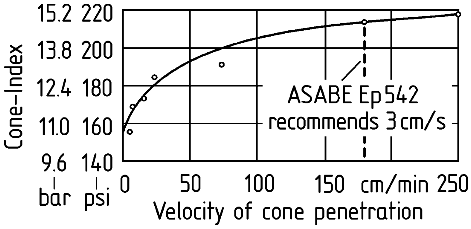


Fig. 2.31: Influence of cone penetration velocity on penetration force [2.49]

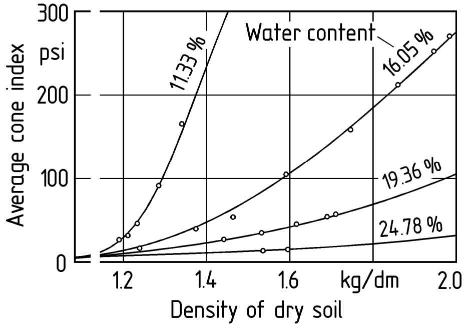


Fig. 2.32: Influence of water content (% of dry bulk volume) on cone penetrometer forces as measured in a compaction cylinder [2.50]

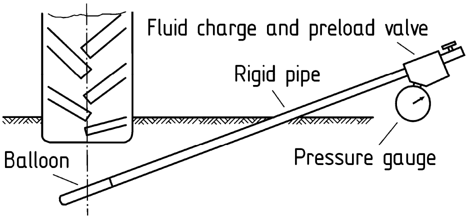


Fig. 2.33: The hydrostatic *Bolling* Sensor [2.52]

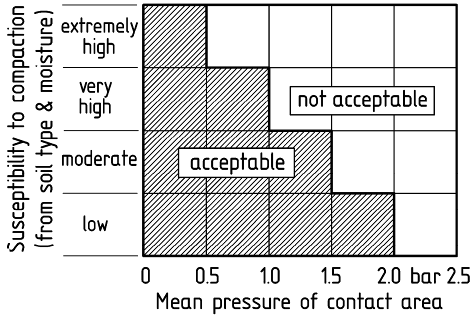
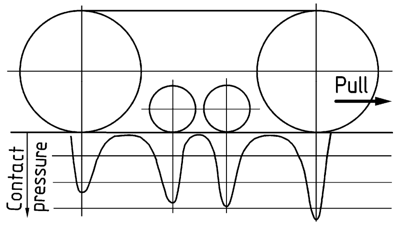


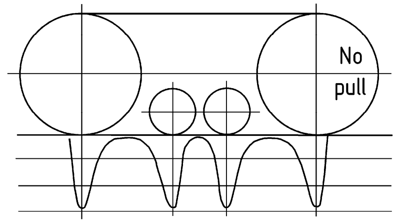
Fig. 2.34 (right)**:** Preventing harmful soil compaction following VDI guideline 6101 [2.53]

2.4 Traction and ground pressure of belt tracks

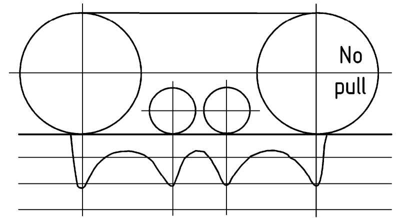
High-power wheel tractors can approach limits regarding tire dimensions and space to install them. Four-wheel drive has been a solution; typically in Europe with different diameters front and rear. In North America, Australia and other countries, equal diameters are popular too, often with duals.



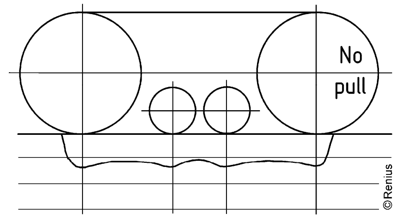
Solid dry loam soil, moderate belt tension, pull force not balanced by front ballast



Solid dry loam soil, moderate belt tension



Solid dry loam soil, very high belt tension



Moist soft sand soil, normal belt tension

Fig. 2.36: Local contact pressures under belt drives (full tracks), influence of soil strength belt tension, and pull (from [2.53], modified)

Tractors with iron track laying systems were used in the 1940s and 1950s; for example in Russia and China. Their main disadvantage is their poor ability on the road.

The Caterpillar "Challenger" entered the market in 1987, overcoming this handicap with rubber belts. The tractor achieved a certain worldwide success making rubber belts popular in general.

Research has been done to compare belt traction potential and soil stresses with four-wheel drives [2.22, 2.55].

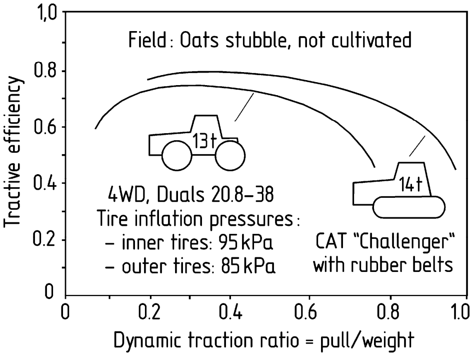


Fig. 2.35: Belt drive versus 4WD [2.55]

Fig. 2.35 [2.55] shows results for a typical uncultivated stubble field. In most cases, the Challenger offers a higher maximum pull/weight ratio and higher tractive efficiency. Practical experience in Europe was able to confirm this; however, with some concerns in the case of thin loose layers on hard surfaces, short heavy rains, and fields with sharp stones or slopes.

Ground pressures, as calculated from weight and contact area, are misleading [2.56–2.58, 2.53], as there are pressure peaks below the track rollers, **Fig. 2.36**.

Belt drives develop their benefits mainly under very soft soil conditions and high belt tensions so that the belt sections between the road wheels become carrying. Well adjusted ballast is recommended.

2.5 Mechanics of a pulling tractor

A rear-wheel drive tractor is simplified to a plane rigid body without acceleration in a horizontal operation without slope, but with pulling force under angle α, **Fig. 2.37**. The "productive" implement pull force is

$$F_P \cdot \cos\alpha = F_T - F_{R,f} \qquad (2.21)$$

Calculation of F_T and $F_{R,f}$ needs the actual vertical axle loads $F_{W,f}$ and $F_{W,r}$ as

$F_T = \kappa_r \cdot F_{W,r}$ and $F_{R,f} = \rho_f \cdot F_{W,f}$

It may be noted that $\rho_f = f_f / r_f$ (2.22)

If we assume that $F_{R,f}$ is in line with F_T (offset neglected), rear axle load from the moment equilibrium related to point A is:

$$F_{W,r} = \frac{F_W(e+f_f)+F_P\cdot\cos\alpha\cdot h+F_P\cdot\sin\alpha(a+b+f_f)}{a - f_r + f_f}$$

- equation (2.23)

Using the moment equilibrium at point B:

$$F_{W,f} = \frac{F_W(a-f_r-e)-F_P\cdot\cos\alpha\cdot h-F_P\cdot\sin\alpha(b+f_r)}{a - f_r + f_f}$$

- equation (2.24)

The following equation can be recommended for calculation control:

$$F_W = F_{W,f} + F_{W,r} - F_P \cdot \sin\alpha \qquad (2.25)$$

$F_P \cdot \sin\alpha$ and $F_P \cdot \cos\alpha$ increase the rear axle load and decrease the front axle load. This is usually favourable in the case of rear 2WD, but front ballast can be necessary for adequate steering safety.

Rear axle torque can be expressed by

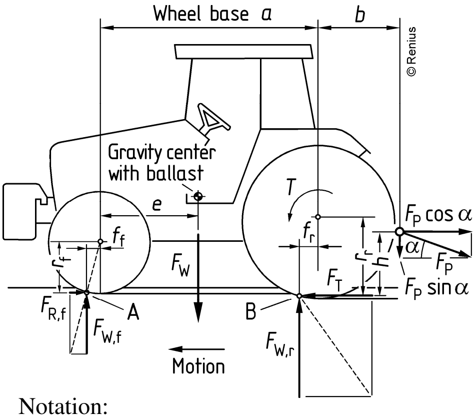
$$T = (\kappa_r + \rho_r) \cdot F_{W,r} \cdot r_r \qquad (2.26)$$

The moment arm r_r is usually a few percent less than the rolling radius [2.59].

Typical results of 2WD tractor pull mechanics are demonstrated by **Fig. 2.38**.

Four-wheel drive can considerably increase pull and efficiency. Productive implement pull is equal to

$$F_P \cdot \cos\alpha = \kappa_f \cdot F_{W,f} + \kappa_r \cdot F_{W,r} \qquad (2.27)$$



Notation:

Index f means front and index r means rear,
a, b, e, f, h, r and α as shown in the drawing.
F_P = total implement force
F_W = tractor weight force with ballast
$F_{W,f}$ and $F_{W,r}$ = axle loads, when pulling
$F_{R,f}$ = front axle rolling resistance
F_T = rear axle net traction
T = rear axle driving torque

Fig. 2.37: Pulling tractor with rear-wheel drive. Term *e* calculated from static axle loads

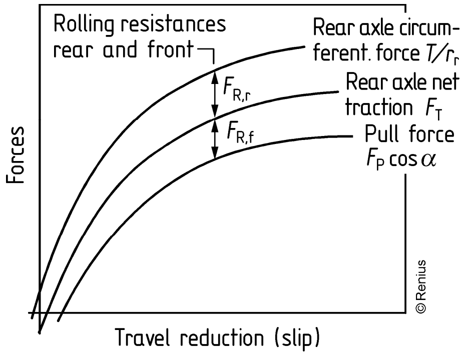


Fig. 2.38: Performance curves of a 2WD tractor

With the assumption $(\kappa_f + \rho_f) = (\kappa_r + \rho_r)$ (*Holm*-equation 2.18), the relation of axle torques becomes very simple:

$$\frac{\text{front axle torque}}{\text{rear axle torque}} = \frac{F_{W,f} \cdot r_f}{F_{W,r} \cdot r_r} \qquad (2.28)$$

2.6 Differential mechanics and average axle slip

Axle differential gears allow different wheel speeds on the left and right.

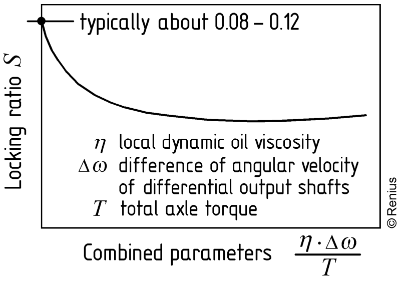


Fig. 2.39: Natural differential locking ratio [2.61]

Ideal torque balancing is usually superimposed by friction between rotating elements resulting in a certain lock-up, which can be addressed by the **locking ratio** S, initially proposed by ZF [2.60]:

$$S = \frac{\text{Torque difference}\,|\text{left-right}|}{\text{Sum of torques left and right}} \qquad (2.29)$$

Factors affecting S are viscosity, sliding speed, and load, which can be combined to a characteristic term similarly to fundamentals of the journal bearing theory, **Fig. 2.39**.

The highest locking effect appears when there is no internal sliding.

Calculating average axle slip is slightly complicated. A possible derivation is shown in **Table 2.4**.

Table 2.4: Calculation of average axle slip in the case of given wheel slips on the left and right

Notation for a driving axle with different slip values on the left and right:

- Slip at left wheel	i_{left}
- Slip at right wheel	i_{right}
- Circumferential speed left wheel	$v_{\text{o,left}}$
- Circumferential speed right wheel	$v_{\text{o,right}}$
- Forward speed left and right with wheel with slip	v
- Tractor speed with slip	v
- Tractor speed with zero slip	v_{o}

Basic equations:

1. Derived from basic differential mechanics: $v_{\text{o,left}} + v_{\text{o,right}} = 2\,v_{\text{o}}$

2. Slip mechnics left wheel: $v = v_{\text{o,left}}(1 - i_{\text{left}}) \longrightarrow v_{\text{o,left}} = \frac{v}{1 - i_{\text{left}}}$

3. Slip mechanics right wheel: $v = v_{\text{o,right}}(1 - i_{\text{right}}) \longrightarrow v_{\text{o,right}} = \frac{v}{1 - i_{\text{right}}}$

Average axle slip $i_{\text{m}} = \frac{v_{\text{o}} - v}{v_{\text{o}}}$, with equation 1: $i_{\text{m}} = \frac{v_{\text{o,left}} + v_{\text{o,right}} - 2v}{2\,v_{\text{o}}}$

Equations 2 and 3 introduced, fraction cancelled by v, and some further transformations lead to:

$$\boxed{i_{\text{m}} = 1 - \frac{2\,(1 - i_{\text{left}})\,(1 - i_{\text{right}})}{2 - i_{\text{left}} - i_{\text{right}}}} \text{ ——— Equation (2.30)}$$

Example: Heavy plowing after some rain without differential lock:
Slip right wheel (in a dry furrow) = 10%, left wheel (wet surface) = 30%
Average slip i_{m} = 21.25% (not 20%!)

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2.7 All-wheel drive versus two-wheel drive

Driving of both axles (4WD) improves off-road traction, traction efficiency, and front end loading productivity. It also enables larger front tires and 4WD braking solutions (chapter 3). In Europe, its breakthrough was from 1975 to 1990.

Performance of 4WD is sometimes underestimated in spite of early recognized benefits, for example by *Söhne* [2.21].

A paper of the author specified the advantages through modelling and field measurements at Deutz tractors, **Table 2.5** [2.37].

Pull force at 20% slip was 42.8% higher for the 4WD although the additional use of vertical axle load for traction was only 25%.

This disproves the rule of sceptical farmers that the traction gain by 4WD is only equal to the additionally used axle load: *"Hence, buying a larger 2WD tractor for the same money"* usually makes no sense.

Reasons for the overproportional pull gain and higher tractive efficiency are the direct compensation of rolling resistance at the front axle and the multi-pass effect at the rear axle. Heavy operations of 4WD tractors can generate the so-called "power hop". Fundamentals and strategies against it are outlined in [2.62].

Table 2.5: Four-wheel drive versus two-wheel drive, calculations for heavy pulling off-road [2.37]

Specifications:	Two-wheel drive	Four-wheel drive
- Nominal engine power	75 kW	75 kW
- Tires front/rear	11.00-16/18.4-38	14.9-24/16.9-38
- Tractor net weight	46 kN	50 kN
- Total vertical load (ballast)	60 kN	60 kN
- Vertical load front/rear	20% / 80%	30% / 70%
- Travel reduction (slip)	20%	20%
- Multi-pass effect	not relevant	$\kappa_f + \rho_f = \kappa_r + \rho_r \quad \kappa_r \cong 1.1\,\kappa_f$
- Cultivating stubble, average soil moisture		(equations 2.18 and 2.19)

2WD

Net traction rear

$$F_{T,r} = \kappa \cdot F_{W,r} = 0.4 \cdot 48\ \text{kN} = 19.2\ \text{kN}$$

Rolling resistance front

$$F_{R,f} = \rho_f \cdot F_{W,f} = 0.1 \cdot 12\ \text{kN} = 1.2\ \text{kN}$$

Pull force (horizontal)

$$F_P = F_{T,r} - F_{R,f} = 19.2 - 1.2 = \mathbf{18\ kN}$$

Rolling resistance rear

$$F_{R,r} = \rho_r \cdot F_{W,r} = 0.1 \cdot 48\ \text{kN} = 4.8\ \text{kN}$$

Tractive efficiency

$$\eta_T = \frac{F_P}{F_P + F_{R,f} + F_{R,r}} \cdot (1 - i) = \frac{18.0\ \text{kN}}{24.0\ \text{kN}} \cdot (1 - 0.2)$$

$$= 0.75 \cdot 0.80 = \mathbf{60.0\ \%}$$

4WD

Net traction rear

$$F_{T,r} = \kappa_r \cdot F_{W,r} = 0.44 \cdot 42\ \text{kN} = 18.5\ \text{kN}$$

Net traction front

$$F_{T,f} = \kappa_f \cdot F_{W,f} = 0.4 \cdot 18\ \text{kN} = 7.2\ \text{kN}$$

Pull force (horizontal)

$$F_P = F_{T,r} + F_{T,f} = 18.5 + 7.2 = \mathbf{25.7\ kN}$$

Rolling resistance front

$$F_{R,f} = \rho_f \cdot F_{W,f} = 0.1 \cdot 18\ \text{kN} = 1.8\ \text{kN}$$

Rolling resistance rear

$$F_{R,r} = \rho_r \cdot F_{W,r} = 0.06 \cdot 42\ \text{kN} = 2.5\ \text{kN}$$

Tractive efficiency

$$\eta_T = \frac{F_P}{F_P + F_{R,f} + F_{R,r}} \cdot (1 - i) = \frac{25.7\ \text{kN}}{30.0\ \text{kN}} \cdot (1 - 0.2)$$

$$= 0.857 \cdot 0.800 = \mathbf{68.6\ \%}$$

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2.8 High-power traction drives: Alternative systems

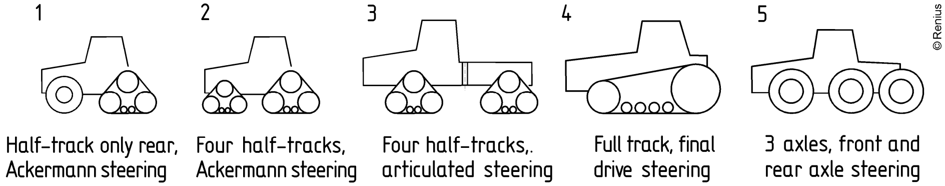
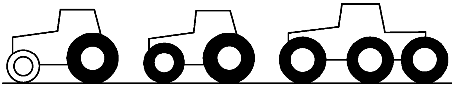

1
2
3
4
5
Half-track only rear, Ackermann steering
Four half-tracks, Ackermann steering
Four half-tracks,. articulated steering
Full track, final drive steering
3 axles, front and rear axle steering
© Renius

Fig. 2.40: Five alternatives for high-power tractors including the application of modern belt drives

Two-axle standard tractors approach limits at 350–400 kW regarding necessary tire sizes and axle loads. Alternative drive systems are listed in **Fig. 2.40**: A suspended half-track tractor (concept 1) was presented around 1990 as a prototype by the National Institute of Agricultural Engineering (NIAE), Silsoe, with encouraging results. Fitted to the rear axle of a 75 kW Ford tractor, each rubber belt unit offered a contact area of 400 x 1027 mm. Case IH and CLAAS recently presented similar designs. A tractor of concept 2 was announced by John Deere in 2019 while design 3 was introduced in series by CASE IH in 1996, later similar by John Deere.

Concepts 1–3 have the advantage that the steering structures can remain identical with those of wheeled versions while final drive gear ratios can be reduced.

A common disadvantage of track systems is, in some markets, the limited ability for on-road driving and transport.



	2WD	4WD	Three-axle drive
Typical tractive efficiency	60 %	69 %	73 %
Typical gain	0 %	15 %	22 %

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Fig. 2.41: Benefits of two or three driving axles

Most such tractors are able to run at 40 km/h but top speeds of 50 or even 60 km/h are also required, mainly in Europe.

Another disadvantage is the high fuel consumption on the road, mainly of full tracks.

While rolling resistance is pretty low under field conditions, it is relatively high on the road (as compared to wheels). Full tracks have in addition to the contact rolling resistance relatively high losses due to the rubber track preload and the steering system.

Full track tractors are, in the author’s opinion, machines for high pulling forces on soft soils – typically including heavy off-road transports; for example, for sugar cane, cereals (from combines) or slurry.

Initial costs are higher per engine hp than for 4WD standard tractors. This can pay in the case of intensive use; for instance, working at least 1000 h per year.

Regarding Europe, tractors with more than two axles are studied by Deutz-Fahr and Fendt as an alternative as they offer excellent on-road qualities, can meet public road regulations without problems, and may not be more expensive than full track concepts at almost the same performance.

This was the reason that AGCO-Fendt developed the TRISIX, a study prototype of design type 5 of Fig. 2.40 [2.63, 2.64].

Modelling of traction mechanics by the author [2.63] had pointed out that a third axle increases performance due to the multi-pass effect at two axles, **Fig. 2.41**.

2.9 Mechanics of overturning

Working on slopes. Slopes can limit tractor missions [2.65-2.68]. Criteria for operations on hilly fields are:

- Tractor steering concept [2.65]
- Tractor track width
- Tractor gravity center [2.68]
- Contact between tires and surface (soil type, wheel type, side forces, drift angles, high or low traction)
- Influence of implement masses
- Working speed and turning radius

Tractor overturning is a very important source of accidents, as indicated by statistics. **Fig. 2.42** demonstrates the mechanics of overturning sideways (without implements) with the pivot position P, the gravity center of the front axle S_{FA} and of the tractor S.

Determination of S: See OECD code 2.

In the first stage of overturning sideways, the front axle remains in contact with the ground. Therefore, an auxiliary gravity center S' is calculated (excluding front axle) using S and S_{FA} and the connected masses. A straight line through P and S' delivers the auxiliary point H. Instability exists if the force directions of F'_W (tractor weight minus front axle weight) or of the rear axle load $F_{W,r}$ are within the plane PHB.

Fig 2.43 shows the critical slopes for all travel directions of a European standard tractor, calculated by *Schwanghart*.

The most critical slope angles are found for driving directly uphill (90°) and along a "contour line" (at 0 and 180°), while all downhill directions are less critical.

Static slope limits are, however, not valid for most practical tractor operations, as dynamic effects such as hillside obstacles or uneven surfaces can reduce the slope limits considerably. As a rough approximation, dynamic slope limits are about 50–60% lower than static values, high values for small tractors due to the poor suspension effects of the tires). A reasonable limit can be seen at slopes of about 20° (=36 %) as already mentioned in chapter 2.1.

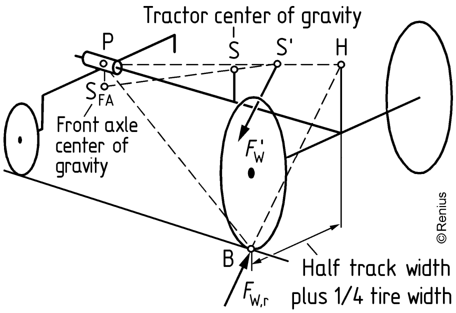
Tractor center of gravity
P
S
S'
H
S_{FA}
Front axle center of gravity
F'_W
B
$F_{W,r}$
Half track width plus 1/4 tire width
©Renius

Fig. 2.42: Critical state for static sideways overturning of tractors with pivoted front axle

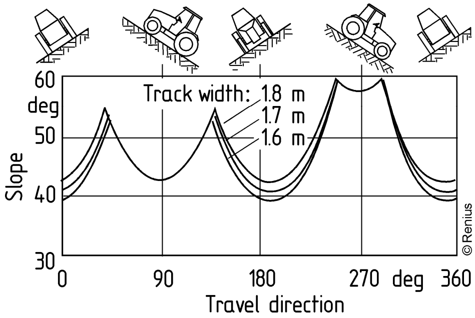

Fig. 2.43: Field slopes as critical for static overturning of a standard tractor, Deutz-Fahr DX 4.70 (66 kW, 1987), calculated by *Schwanghart*, based on his fundamentals in [2.69]

For this reason, all lubrication systems and hydraulic circuits of a standard tractor should be able to handle a permanent tractor tilt angle of at least 20°.

In the case of meadows with slopes much greater than 20°, special tractors have been developed with lower centers of gravity and duals realizing about 30°.

Implements and trailers can further reduce critical slopes, mainly in the case of front loaders in up-positions.

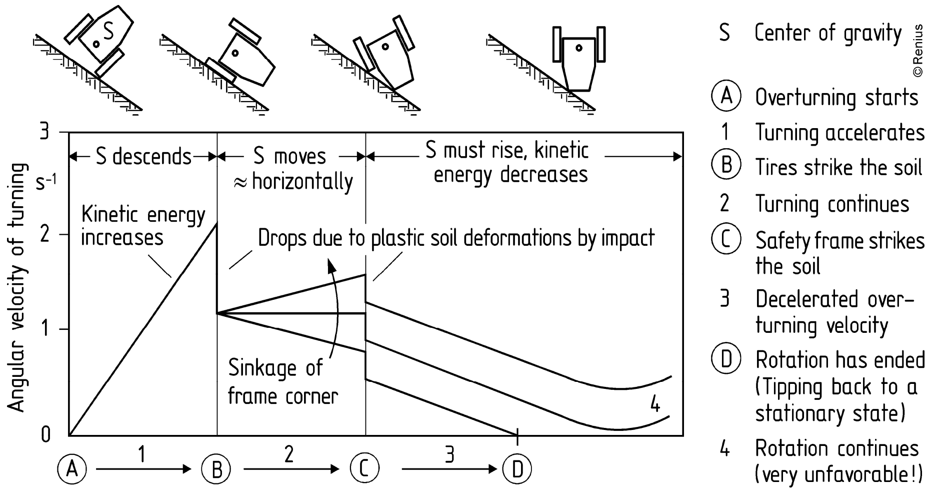

Fig. 2.44: Simplified analysis of sideways overturning of a standard tractor with a safety frame on a slope of 1:2.5 (40%) by showing the energy balance in typical phases. Angular velocities are shown for a medium sized tractor. Calculation strategy of *Schwanghart* [2.69]

Overturning backwards can happen when driving uphill with heavy pulling and a high line of draft; for example, with trailers or accelerating with heavy mounted implements in up position. It is very dangerous, as the driver may not be able to react fast enough. Calculations can be done on the basis of equation (2.24).

Overturning sideways on a slope in the field is complicated to calculate as the tractor is moving in three dimensions and its impacts with the ground are both elastic and plastic. Mathematical models have been developed by several authors to simulate this type of overturning [2.65].

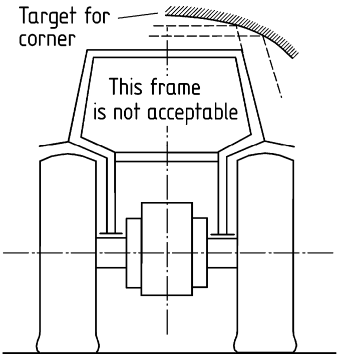

Fig. 2.45: Target for safety frame corner preventing continuous overturning [2.70]

Experiments have been undertaken by *Schwanghart* at TU Munich under *Söhne* to investigate the mechanics of overturning sideways. The applied strategy was to observe the energy balance and dissipations (due to ground contact) during the overturning, **Fig. 2.44**. Kinetic energy increases when the center of gravity decreases, and vice versa. Mechanical energy is reduced by ground contacts due to plastic impacts. For related tests, see chapter 6.7.

It is one important objective of this calculation to investigate whether the tractor rotation is ending (phase 3) or not (phase 4). When the rotation continues (4), damage to the tractor and risks for the driver increase.

The result is influenced by the position of the edge of the structure, **Fig. 2.45**.

Computer programs can plot a target line [2.69] and prove whether the rotation ends or not (mandatory within the EU for narrow track tractors T2).

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3. Chassis design

3.1 Tires and growth limits

For global requirements, see **Table 3.1**. Tractor tires must be able to perform not only off-road but in many countries also on-road. International top speeds are often 28-32 km/h, which is about 20 mph.

Table 3.1: Requirements on traction tires

- To carry required loads on-road and off-road
  - vertical: weight, ballast, implement forces
  - lateral: side forces at steering and on slopes
  - longitudinal: pulling and braking forces
- To offer high and efficient off-road traction
  - by high net traction ratios
  - by large contact area / low rolling resistance
  - by adequate self cleaning ability
  - by containing fluid ballast if necessary
- To conserve the agricultural soil
  - by adequate profile and large contact area
  - by enabling low off-road inflation pressures
  - by a uniform contact pressure distribution
- To offer adequate on-road properties
  - by sufficient load capacity (homologation)
  - by high maximum speed (homologation)
  - by spring and damping (riding comfort)
- To offer long life, low wear, and low costs

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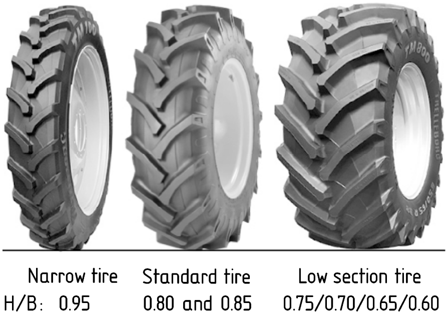

Fig. 3.1: European traction tires for off-road and on-road (courtesy Trelleborg Wheel Systems [3.2]). H section height, B section width

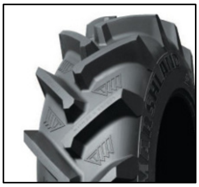

Fig. 3.2 (left): High-lug tires for very soft soils and paddy fields.
Courtesy Trelleborg

In Europe, top speeds increased after World War II from 20 km/h to:

- 40 km/h as EU standard (EU regulations)
- 50 km/h as a popular option
- 60 km/h as an option for large tractors

Different designs are used:

- Bias ply tires with inner tube
- Radial ply tires with inner tube
- Radials and bias tubeless

Traction tires (radials) with standard section size are **coded** as follows [3.1]:

$$\mathbf{\underbrace{20.8}_{\text{Nominal tire width (in)}}\ \underbrace{R}_{\text{Radial ply}}\ \underbrace{38}_{\text{Rim diam.}}\ \underbrace{145}_{\text{Load index}}\ \underbrace{A6}_{\text{Speed index}}}$$

Code for bias ply tires: **20.8-38 145 A6**.

Code for metric width of radial ply tire:

$$\mathbf{\underbrace{520}_{\text{Nominal tire width (mm)}}/\underbrace{70}_{\text{Section code}}\ \underbrace{R}_{\text{Radial ply}}\ \underbrace{38}_{\text{Rim diameter}}\ \underbrace{145}_{\text{Load index}}\ \underbrace{A6}_{\text{Speed index}}}$$

Section code (also "aspect ratio" [3.1]) is obtained by dividing section height H by section width B, **Fig. 3.1**.

Narrow tires are popular for row crop operations, height-to-width ratio often 95.

Standard tires have height-to-width ratios of 0.80 or 0.85 and are produced in high volumes. Example: 18.4R34 (80 hp tractor).

Low section tires have a height-to-width ratio below 0.80 down to 0.60.

High-lug tires are necessary for very soft soils and paddy fields, **Fig. 3.2**. Their performance on dry upland is poor.

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The two basic design principles of traction tire carcasses are shown in **Fig. 3.3**.

Radial ply tires or “radials” have extra-flexible sidewalls enabling a larger soil contact area and a slightly more uniform transfer of the traction forces to the soil and to the rim as well. The benefits are lower contact pressure, higher net traction, lower rolling resistance and rut depth, higher tractive efficiency, and longer service life. Benefits should not be overevaluated. This was the reason that the traction ratio κ versus slip i was increased in Fig. 2.13 by only about 10–12% for off-road, as compared with former published values for *bias ply* tires.

The main disadvantage is the higher price, in a second order also a higher vulnerability of the flexible, thin flanks. Nevertheless, radials have become very popular, mainly in high-tech markets.

When traction performance is not in the foreground, as for example for narrow tires, bias tires can remain sufficient [3.2].

Both concepts are available either with tubes or tubeless (trend: no tubes).

Tire load capacities are listed in the tire manufacturers’ brochures, **Table 3.2** [3.2]. They depend on:

- Carcass design (ply rating)
- Inflation pressure
- Maximum working speed
- Torque load level (only at low pressures)

“Very high flexion tires” (ECE R 106) have a special carcass allowing higher loads above 10 km/h; see, for example, the Bridgestone VF tire line.

Popular tires are listed in **Table 3.3**.

Sizes with the same SRI can also be classified in groups by the “**R**olling **C**ircumference **I**ndex” RCI.

Increasing tractor power and weight requires larger tires. *An increase of width is usually cheaper than a larger diameter because of both, tire and transmission costs.*

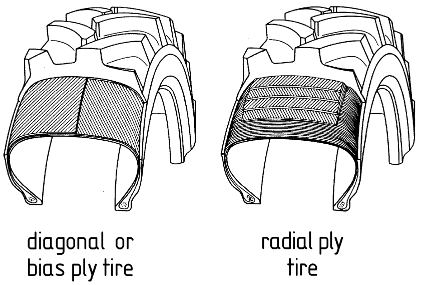


Fig. 3.3: The two basic carcass designs for traction tires: bias ply and radial ply [3.1–3.3].

Table 3.2: Maximum loads for tire 600/65R38 as a function of inflation pressure and maximum speed [3.2]. Typical rear tire of a 120 kW tractor

Inflation bar	0.6	0.8	1.0	1.2	1.6	2.1
pressure psi	9	12	14	17	23	30
Max. loads kg	|	|	|	|	|	|
65 km/h (D)	2125	2455	2785	3075	3660	---
50 km/h (B)	2230	2575	2925	3230	3845	---
40 km/h (A8)	2325	2690	3050	3370	4010	---
30 km/h (A6)	2445	2825	3200	3540	4210	---
10 km/h (A2) low torque	2890	3335	3785	4185	4980	6225
10 km/h (A2) high torque	2445	2825	3200	3540	4210	

speed index SI

Table 3.3: Selected traction tires (radials) with common SRI. Courtesy Trelleborg [3.2].
SRI = **S**peed **R**adius **I**ndex = radius for calculation of tractor speeds. ETRTO = **E**uropean **T**yre and **R**im **T**echnical **O**rganisation [3.4]

SRI ETRTO mm	Standard tires H/B=80-85%	Wide series	Extra large series
500	12.4R20	360/70R20	- - -
550	12.4R24	360/70R24	420/65R24
600	14.9R24	420/70R24	480/65R24
625	16.9R24	480/70R24	540/65R24
650	14.9R28	420/70R28	480/65R28
675	16.9R28	480/70R28	540/65R28
700	16.9R30	480/70R30	540/65R30
725	18.4R30	520/70R30	- - -
750	16.9R34	480/70R34	540/65R34
775	18.4R34	520/70R34	600/65R34
825	18.4R38	520/70R38	600/65R38
825	20.8R34	- - -	710/60R34
875	20.8R38	580/70R38	650/65R38
925	20.8R42	580/70R42	650/65R42
975	20.8R46	710/70R42	900/60R38
1025	- - -	710/75R42	900/60R42
1125	- - -	- - -	900/65R46

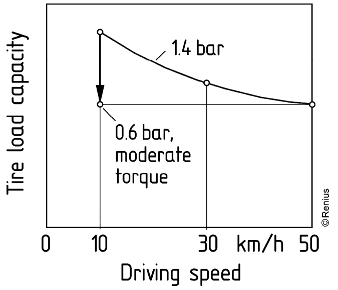


Fig. 3.4: Typical potential of inflation pressure reduction at low speeds without capacity losses

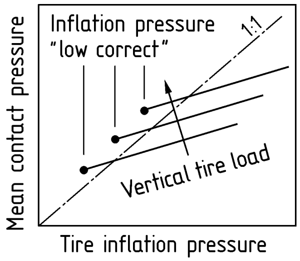


Fig. 3.5: "Low correct" tire inflation pressure for a given tire. Source: [3.6] modified

International standards have been developed for tractor tires and similar mobile machinery:

- ISO 4223-1 Definitions for pneum. tires
- ISO 4251-1 Tires (PR rated) and rims
- ISO 4251-2 Load ratings for bias tires
- ISO 4251-3 Rims
- ISO 4251-4 Tire classification code
- ISO 4251-5 Log skidder tires
- ISO 7867-1 Metric agric. tires and rims
- ISO 7867-2 Metric agric. tires, loads
- ISO 8664 Radials, load index, speed symbol

ISO standards are replacing more and more national standards. Additional directives can be important regarding official traffic regulations for driving on public roads.

Improved off-road tire performance is possible by reduced tire inflation pressure, as specified loads usually increase with lower speeds at constant inflation, **Fig. 3.4**.

The **"low correct" inflation pressure** is the lowest specified inflation for a given speed and tire load [3.5-3.7], **Fig. 3.5**, offering the lowest mean contact pressure.

Tire dynamics influence tractor dynamics.

Some general trends are published in [3.8-3.11] and summarized in **Fig. 3.6**. The two far left diagrams address stiffness and the right ones damping. Inflation pressure influences the tire stiffness. Radials are less stiff than bias (favourable) but offer less damping (not favourable). Damping increases off-road (for both types) due to additional soil damping [3.12]. Aging effects influence tire properties [3.11]. For more details and references see chapter 6.5.

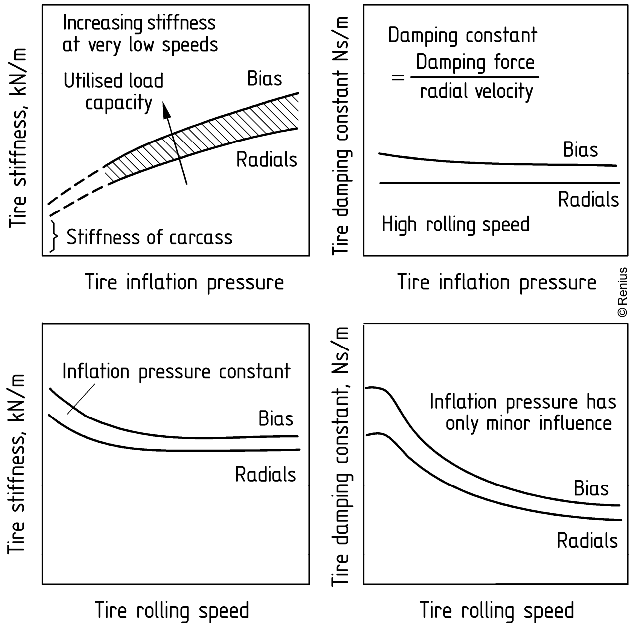


Fig. 3.6 (left): Stiffness and damping of bias and radial ply tires, general trends [3.8]

The **nominal tire loads** are a strong limit for homologated axle loads of the whole tractor influencing basically the maximum payload which is important for:

- Mounted front implements
- Mounted rear implements
- Vertical linkage and drawbar loads
- Container for seeds, fertilizer, chemicals
- Ballast

Tractor payload should be 60–80 % of the tractor net weight to meet these needs – lower values for very large tractors can be tolerated (no heavy mounted implements).

Compliance with the maximum nominal tire loads is in many countries a must for driving on public roads to meet legal limits, but is recommended also for driving off-road because of reliability reasons.

Limits to tire growth result from the fact that the tire dimensions must increase overproportionally with tractor dimensions due to a similarity law, which was first addressed by *Galileo Galilei* (1564-1642) [3.13].

The key point is that the enlargement of a system by a factor λ increases the weight by λ^3 (in the case of same materials) but increases all sections of weight supporting elements only by a factor of λ^2.

Such a supporting section is the tire-soil contact area of a tractor as noted by *Söhne* [3.14] and other experts of terramechanics.

If the tire dimensions are enlarged by a factor λ, the contact pressure increases by the factor $\lambda^3/\lambda^2 = \lambda$ and this means:

The tire dimensions must be enlarged by a factor of $\lambda^{1.5}$ *to keep the average contact pressure between soil and tire constant.*

The tractor wheel base has been identified to be a very good geometrical size indicator for studies of this kind [3.15, 3.16]. Regarding **Fig. 3.7**, the net weight corresponds with an exponent of 2.89 (theory: 3.0). A similar analysis from *Rempfer* [3.16], based on other tractor models, resulted in an exponent of 3.11.

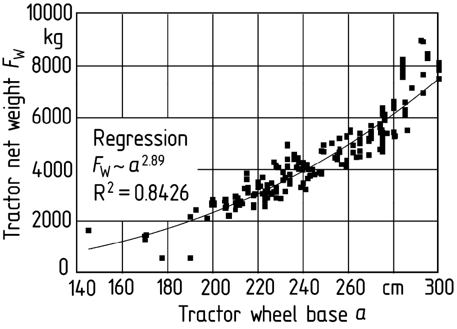


Fig. 3.7: Tractor weight versus wheel base [3.15]

The growth limits can be mitigated by tire inflation control systems (Table 2.1). Design fundamentals have been presented by *Rempfer* [3.16]. Fendt (AGCO) was the first company to offer a factory-installed integrated system for tractors (protos 2011, series 2013 for Vario 800 and Vario 900 tractors).

A practical tire selection should mainly consider

- Tire loads (for maximum tractor weight)
- Performance (mainly off-road)
- Soil compaction (off-road)
- Tractor width limits (public roads)
- Tractor dimensions (clearances, duals)

Regarding soil contact power density, the following *power load index* can be useful:

$$\frac{\text{Nom. PTO kW}}{\Sigma\,(\text{b}\cdot\text{D})\text{ powered}} = 50 \ldots 65\ \frac{\text{kW}}{\text{m}^2} \qquad (3.1)$$

PTO power below 120 kW

High power, high speeds, high axle loads (ballast), duals at rear axle, low inflation pressures

where

b [m] = tire width

D [m] = tire diameter.

Based on [3.17, 3.18], the projection $b \cdot D$ and the tire load can be used defining a

$$\text{Contact pressure index} = \frac{\text{tire load}}{\text{b}\cdot\text{D}} \qquad (3.2)$$

A limit of 0.25 bar (3.6 psi, 2.5 N/cm^2) can be recommended for soft soils.

3.2 Chassis concepts

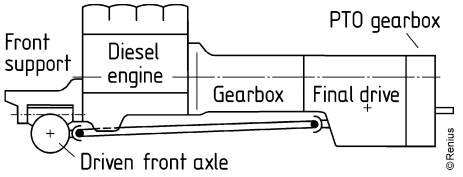


Fig. 3.8: Typical block chassis of standard tractors with integrated 4WD

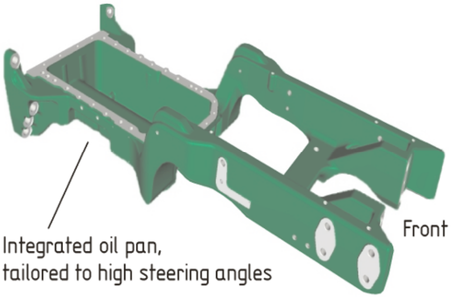


Fig. 3.9: Integration of front support and engine oil pan, John Deere 5R tractor series 2016.
Courtesy John Deere

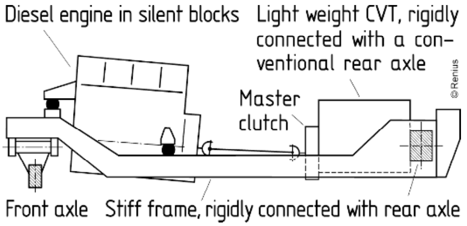

Diesel engine in silent blocks
Light weight CVT, rigidly connected with a conventional rear axle
Master clutch
Front axle
Stiff frame, rigidly connected with rear axle
©Renius

Fig. 3.10: ¾ frame of the Munich Research Tractor, presented 25.02.1988 at TU Munich [3.19]

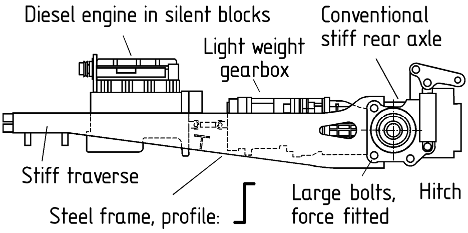

Diesel engine in silent blocks
Light weight gearbox
Conventional stiff rear axle
Stiff traverse
Steel frame, profile:
Large bolts, force fitted
Hitch

Fig. 3.11: ¾-frame of John Deere series 6000, Mannheim, Sept.1992 [3.19]

General requirements on a tractor chassis are:

- To hold the axles, also suspended
- Offer high inner steering angles
- Carry the powertrain
- Carry high payloads front and rear
- Be durable & stiff for bending and torque
- Have low weight and low number of parts
- Be flexible for modifications
- Meet relevant standards and regulations

Payloads for standard tractors should be 60–80 % of the tractor net weight, as mentioned, high values in the case of heavy mounted implements also in front.

Block design (Fordson F 1917) still dominates for standard tractors, **Fig. 3.8**.

High vertical loads in front of the tractor (by front ballast and front implements) can create high bending moments to be carried by the flange which connects the front support with the engine. This critical section requires proper dimensioning and testing. An interesting solution from John Deere simplifies the situation by integrating the front support with the engine oil pan, see **Fig. 3.9**. Tailoring the oil pan to high steering angles gives additional benefits.

Frame concepts are usually not cheaper but can offer interesting advantages:

- Higher component flexibility
- Potential for low noise level
- Lower costs for front hitch/front loader

The GDR standard tractor ZT 300 had a half frame and a soft suspended engine as early as 1967.

The “Munich Research Tractor” was developed to investigate the frame benefits systematically, **Fig. 3.10**. One benefit was the noise reduction, quantified in chapter 6.

John Deere introduced a very successful similar concept in late 1992, **Fig. 3.11**.

Other companies followed with front frames for standard tractors allowing suspended engines; however, not for all their models.

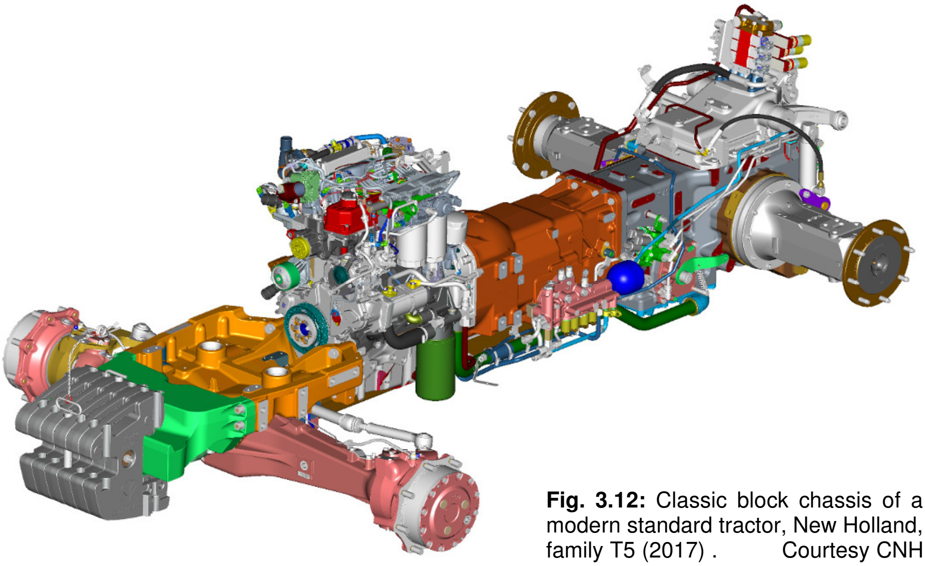


Fig. 3.12: Classic block chassis of a modern standard tractor, New Holland, family T5 (2017) . Courtesy CNH

The following important rule for the design of frames is recommended:

A high torsion stiffness needs closed cross profiles, **Fig. 3.13** [3.20].

Regarding half or ¾ frames (Fig. 3.10 and 3.11) cross profiles can be replaced by a compact front support and a rigid rear axle block. A similar design was successfully developed by Fendt, **Fig 3.14**.

Three design directions can be observed:

- Bolted steel frame (Unimog, CLAAS)
- Welded steel frame (JCB)
- Cast iron frame (Fendt)

Bolted frames offer a high design flexibility, but they need a high number of parts while welding or casting leads to integration. Bolted frames are, for example, applied to frames of the Unimog, **Fig. 3.15**.

A frame allows the soft suspension of the diesel engine in silent blocks, reducing body noise.

A frame may increase the total tractor initial costs *but reduces interface costs for optional front hitch and front end loader*.

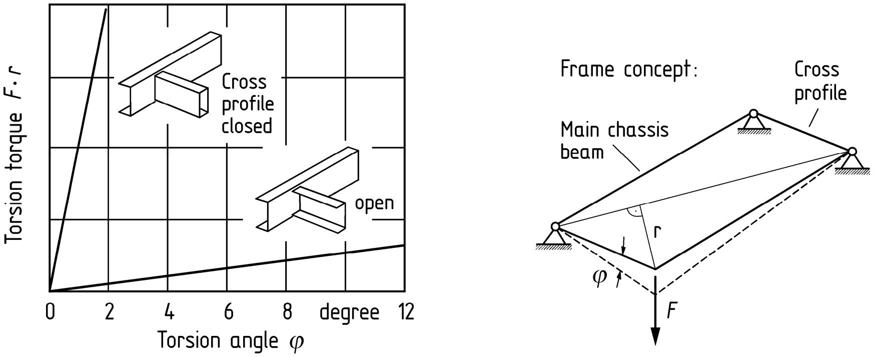


Fig. 3.13: Influence of cross profile design on frame stiffness under torsion. Two closed cross profiles instead of open ones extremely increase torsion stiffness [3.20] – here by a factor of 30!

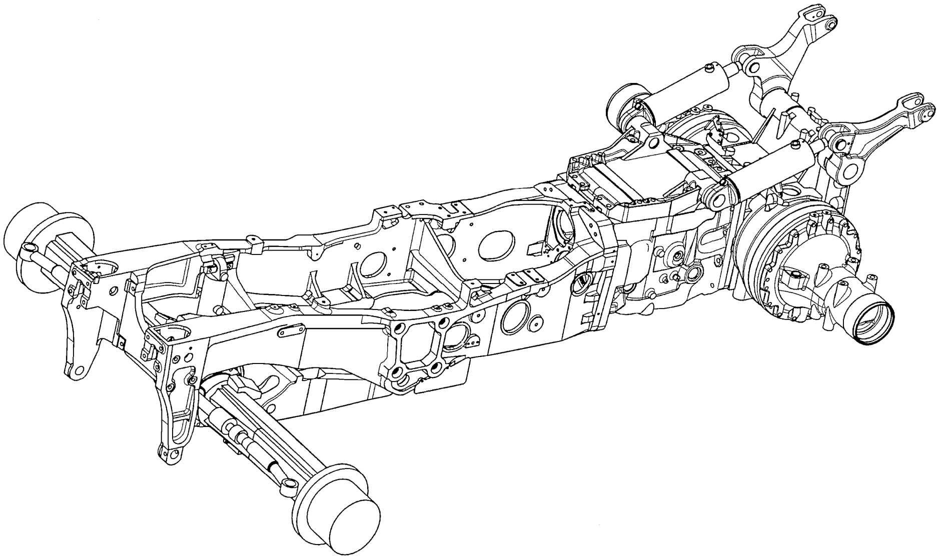


Fig. 3.14: Half cast iron frame (ductile cast iron GGG40) of the Fendt Series 400 Vario, introduced 1999 (later also used for the 500 Vario). Soft engine suspension. Courtesy AGCO-Fendt

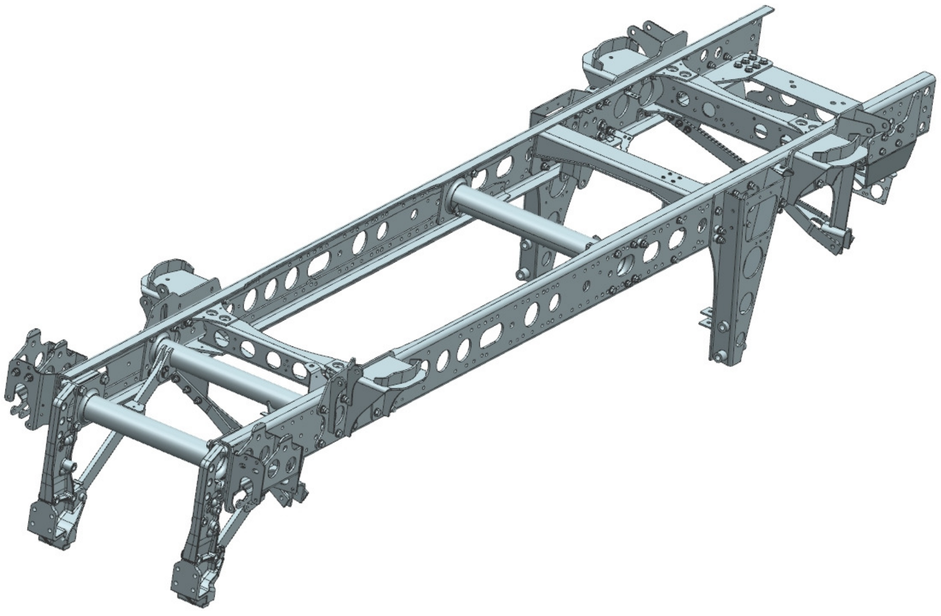


Fig. 3.15: Full steel frame of the Unimog UGN 530 (220 kW). Closed cross profiles for high torsional stiffness, bolted as usual for trucks. Soft engine suspension. As of 2015. Courtesy Daimler AG

3.3 Chassis dimensions and dimensioning

The two **basic chassis dimensions** are

- *Wheel base*
- Lowest possible *track width*

The wheel base is fixed while the track width is usually adjustable; however, with a minimum value, dictated by the width of the cab, the tires, and the tire clearances.

ISO 4252 requires an inside cab width of 900 mm. If we assume small 12.4 inch row crop tires, there is room for a lowest track width of about 1300 mm.

A low **wheel base** results in low turning radii and low initial costs while high values offer more space for the components and better longitudinal stability.

Practical values represent a compromise, for which **Fig. 3.16** may serve as a guideline, representing popular tractor families.

The dimensioning and testing of the chassis structure has to be based on practical loads, in the best case on load spectra.

Critical situations or operations:

- Driving on *uneven roads* with heavy mounted implements front or/and rear
- Heavy mounted *front implements* with large center offset to tractor center line
- Swinging *axle arriving at the stopper* in operations with high front axle loads
- Very heavy *front end loading*
- Driving *down from a high rigid platform* with front mounted implement or ballast
- Driving with a front wheel *against a wall*
- Very high *pulling or braking forces*

These cases result in high **bending and torsion loads**, also simultaneously [3.21].

High bending moments have been observed for front end loading in the rear part of the structure and for heavy front implements or ballast with a critical section at the connection between front support and diesel engine [3.21], **Fig. 3.17** – the reason to apply special, heavy oil pans or frames.

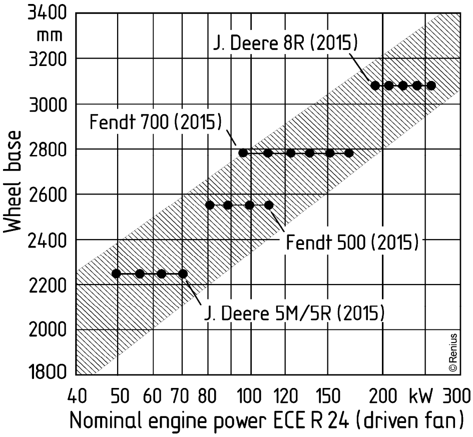


Fig. 3.16: Wheel base guideline for standard tractors with popular tractor families

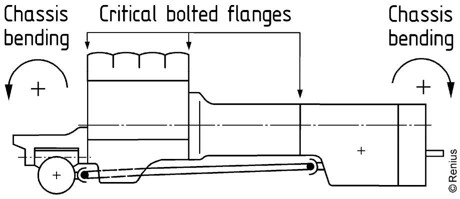


Fig. 3.17: Critical areas of chassis bending

Tractor loads and moments can be defined by three-dimensional coordinates, **Fig 3.18**.

The x-axis is sometimes also chosen in the opposite direction.

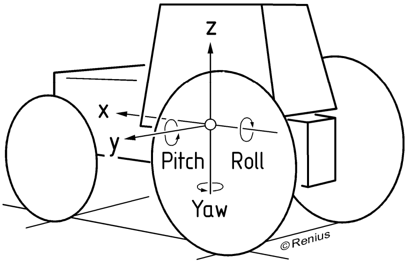


Fig. 3.18: Tractor axis system for vehicle mechanics as defined by ISO 8855 and ISO 2631

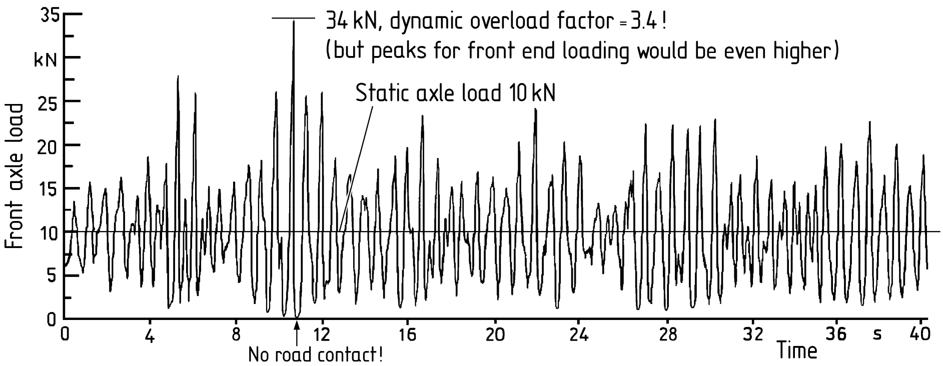


Fig. 3.19: Vertical front axle load of a small tractor moving at 20 km/h on a moderate country road, not asphalted. Rigid front axle. The axle loses contact at 10.7 s on the time scale [3.22]

A rigidly mounted front axle has to carry high dynamic overloads, **Fig. 3.19** [3.22], confirmed by other measurements [3.23]. Lower amplitudes occur for tillage [3.24], but even higher for front end loading [3.25].

Rainflow load cycle counting [3.25- 3.27] is needed for chassis load spectra, **Fig. 3.20**. The chassis bending moment has a positive algebraic sign as defined by Fig. 3.14.

Such a representation also enables the reading of the absolutely highest peak; the red marked column: 23 % + 60/2 % = **53 %**.

High torsional moments are, for example, observed with heavy mounted implements along with large offsets to tractor centerline, on rough terrain or bad roads.

A typical level for the highest torsion moment peak is a value equivalent to rear axle load multiplied by the largest track width (≈ static side overturning moment).

Finite elements methods are recommended to calculate local stress peaks [3.21].

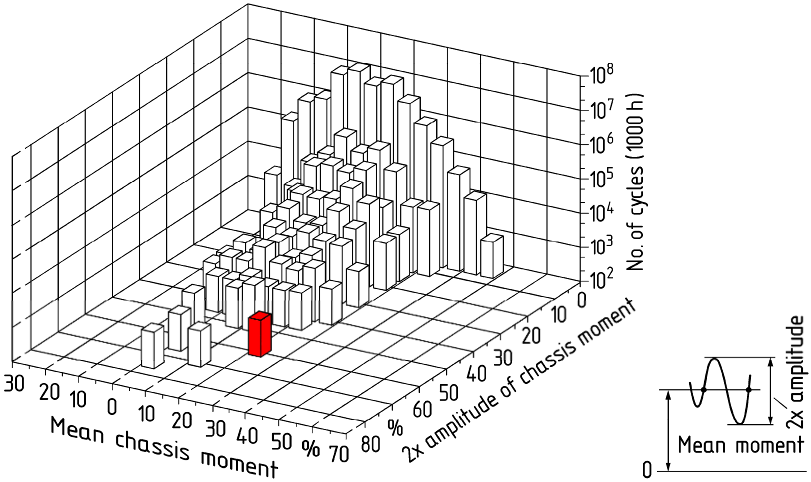


Fig. 3.20: Measured rainflow history of the chassis bending moment in the front part for 1,000 h of mixed tractor operations, suspended axle. 100 % = net tractor weight x wheel base [3.25]

3.4 Four-wheel drive systems

4WD was once the term used for articulated tractors when these tractors had equal wheel diameters front and rear.

This may be one reason why US tractor companies introduced the term “MFWD” Mechanical Front-Wheel Drive for standard tractors. In other countries all tractors with an additional front drive are called 4WD. Its production volume is much higher than that of articulated tractors. Because of these two reasons the following fundamentals concentrate on 4WD systems of standard tractors, **Fig. 3.21**.

In 4WD mode most of them have a *rigid connection between rear and front axle with a typical front axle lead (= circumferential extra speed) of about 1–4 % to improve steering behaviour (chapter 3.8).*

A lead of, for example, 5 % reduces the tractive efficiency off-road by about 1 %.

Type A is a typical first add-on design as, for example, introduced in Europe after World War II as a factory installed option. Extra costs of this first generation were high, about 30 %.

Type B is an improvement regarding the introduction of final drive planetaries. They offer a high reduction ratio (typically 5–8) and allow a smaller diameter of the front differential (for better ground clearance and larger steering angles) and at the same time a higher total reduction ratio, reducing the drive line torques.

Type C was an innovation introduced in 1965 in series by SAME, Italy.

Such an integration can reduce the additional 4WD costs under 20 % also enabling more space aside of the chassis – used, for example, for the fuel tank. Large steering angles are also favoured with C.

Type D seems to be a step back, but became important within the recent decades with the following design advantages:

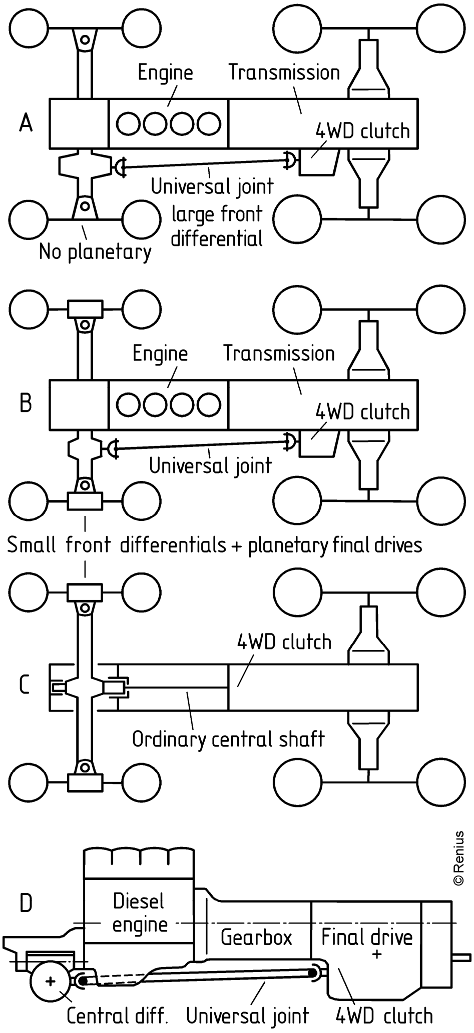


Fig. 3.21: 4WD concepts of standard tractors.
A Typical first optional concept (add-on)
B Improved concept A with planetaries
C Integrated prop shaft, center input
D Integrated prop shaft, offset input

1. No fixed relation of front axle and rear axle center: more freedom for tire selection.
2. Option 2WD easy to realize.
3. Slightly higher maximum inner steering angle for a given chassis contour and a given pendulum angle.

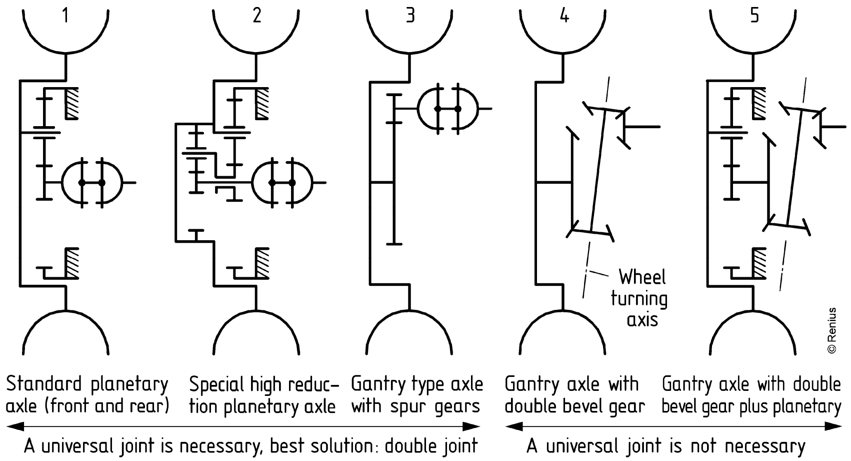

1
2
3
4
5
Wheel turning axis
Standard planetary axle (front and rear)
Special high reduction planetary axle
Gantry type axle with spur gears
Gantry axle with double bevel gear
Gantry axle with double bevel gear plus planetary
A universal joint is necessary, best solution: double joint
A universal joint is not necessary
© Renius

Fig. 3.22: Basic final drives of driven front axles: only one side plotted and differential not shown

Several **final drive concepts** are in use for front axles, **Fig. 3.22**. The most popular one is No. 1 while No. 2 enables higher final drive ratios and No. 3 offers more ground clearance, but is more expensive. Solutions 4 and 5 are typical for Japanese tractors (Yanmar, Kubota from 1976).

Gearing concept No. 1 is today the standard version, **Fig. 3.23** and **3.24**.

The differential ratio is, for example, only 3 (ground clearance) while it is about 5–8 in the planetary unit. Three planetary gearwheels are popular as they center the sun gear automatically. Four planetaries are sometimes used for increased power density but require much design experience (chapter 5.7.4).

"Natural" differential locking ratios (equation 2.29) can be increased by friction plates to achieve locking values for front axles of typically about 45%.

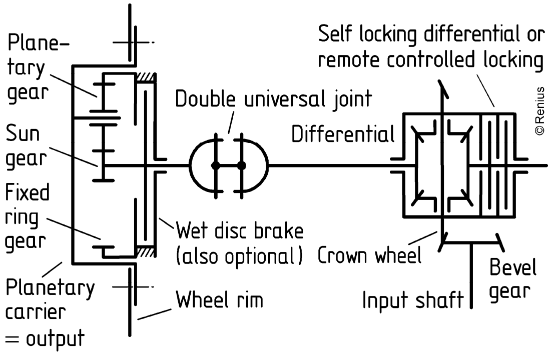


Fig. 3.23 (left)**:** Structure of a typical front axle gearing with outside final drive planetaries and integrated wet disc brakes

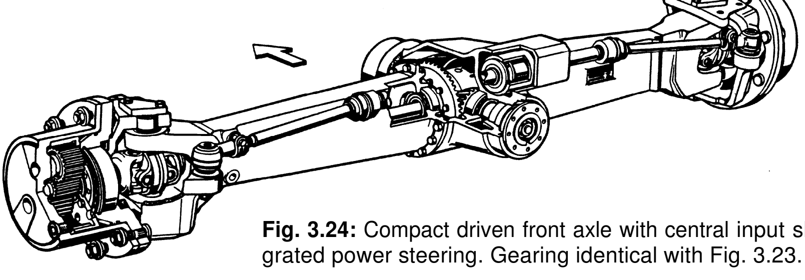


Fig. 3.24: Compact driven front axle with central input shaft and integrated power steering. Gearing identical with Fig. 3.23. Courtesy ZF

3.5 Front axle suspensions

Hydropneumatic suspensions are common for high speeds [3.28-3.32], **Fig. 3.25**.

Cylinder 1 carries the vertical tire load by hydrostatic pressure; accumulator 2 is pre-loaded by hydrogen and hydraulic pressure 3 via valve 4. Damping by orifices 5 and 6 can be variable [3.33]. Position sensor 7 enables feedback to the control box 8. Nitrogen determines the spring characterristics [3.31]. *The polytrophic exponent is more adiabatic than isothermal* [3.34, 3.35]. Design of suspended axles can be planned by **Fig. 3.26** and **Table 3.4** [3.29] – with a marked example. Evaluations are presented in [3.29, 3.36].

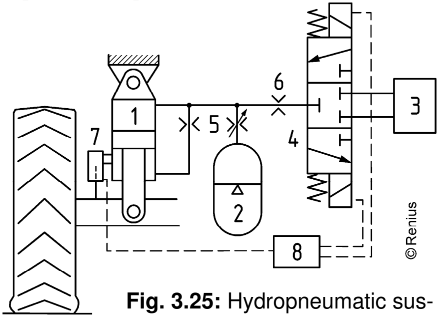


Fig. 3.25: Hydropneumatic suspension, see also [3.30]

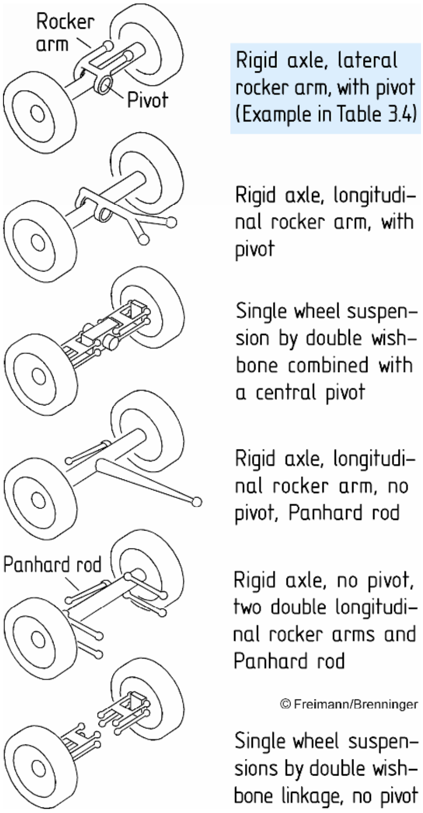

Rocker arm
Pivot
Rigid axle, lateral rocker arm, with pivot (Example in Table 3.4)
Rigid axle, longitudinal rocker arm, with pivot
Single wheel suspension by double wishbone combined with a central pivot
Rigid axle, longitudinal rocker arm, no pivot, Panhard rod
Panhard rod
Rigid axle, no pivot, two double longitudinal rocker arms and Panhard rod
© Freimann/Brenninger
Single wheel suspensions by double wishbone linkage, no pivot

Fig. 3.26: Suspended front axles [3.29]

Table 3.4: Morphologic box for components of hydro-pneumatic suspended front axles [3.29]

Design type Criteria	Rigid axle	Compound axle		Wheel suspension	
Swinging mode	by central pivot		without central pivot		
Rocker arm direction	lateral	longitudinal pushed		longitudinal pulled	
Rocker arm linkage	linked to one side		linked to both sides		
Transfer of lateral wheel forces	by a simple lateral rocker arm	by a stiff longitudinal rocker arm	by a double wishbone	by a Panhard rod	by single wheel suspension linkage
Transfer of longitudinal wheel forces	by a simple longitud. rocker arm	by a stiff lateral rocker arm	by 2 transversal double links	by a special rod	
Transfer of roll moment	no transfer within pendulum clearance	by the suspension springs		by a stabilizer rod	by stabilizer rod and suspension springs
Transfer of pitching moment	no transfer within pendulum clearance	by pushed longit. rocker arm (brake pitching)		by pulled longit. rocker arm (start pitching)	by suspension stiffness control
Blocking of the supension	by oil hydraulics	mechanically	electro-mechanically	no blocking	
Level control	automatically	manually		no level control	

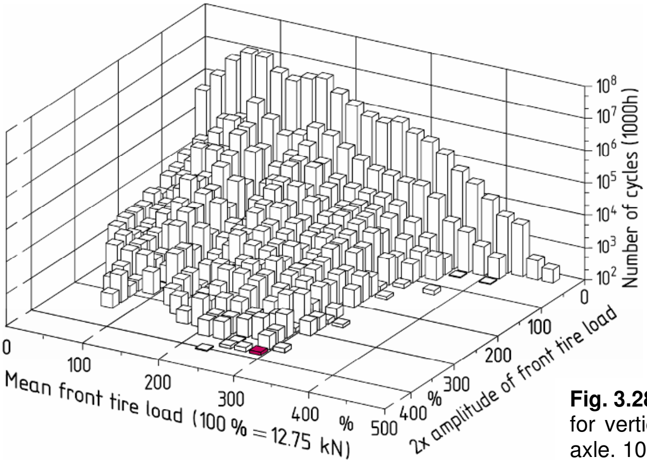
The first factory-installed hydropneumatic *single wheel* front suspension for a produced standard tractor was presented in 2001 by John Deere (series 8000), **Fig. 3.27**. Fendt followed in 2006 with the Vario 936.

General design fundamentals for single wheel suspensions are published in [3.37].

Front axle suspensions and tire inflation control systems can help to prevent the so-called "power hop" [3.38].

Fig. 3.27 (left): Single wheel suspension at front axle, J. Deere 8000, 2001. Courtesy J. Deere

3.6 Axle load spectra and axle efficiencies



Front axle loads including front end loading are classified in two types: Measured *tire loads* by rainflow counting, **Fig. 3.28** [3.25] and *axle torques* by time portions [3.39], **Fig. 3.29** – both to be regarded as guidelines or typical examples.

Fig. 3.28 (above)**:** Rainflow load history for vertical front tire loads, suspended axle. 100% = static tire load = 12.75 kN. All typical operations in Europe for 1000 h. Front end loading is weighted by 13.5% of total time (suspension blocked). Highest peak (the red marked column) results in **480% static load.** Estimated probability 90%. Table of exact cycle numbers are documented in [3.25]

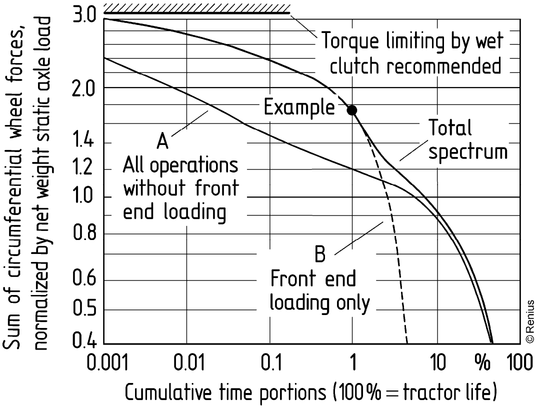


Fig. 3.29 (left)**:** Front axle torque load spectrum for "with and without front end loading", rigid axle [3.39]. Probabilities guessed by *Renius*: For **A** above 95%, for **B** about 50%. Negative torques not considered (important in case of front end loading). Example for reading (•): 1% of time indicates all circumferential axle wheel forces of 175% *and above*

Importance of Fig. 3.28 and 3.29. Both diagrams indicate that the design of a driven tractor front axle *must consider the possibility of a mounted front end loader.*

It is recommended to scale such a diagram not by forces (or torques) such as N or kNm but rather to *normalize* the loads by tractor parameters; in this case, by the static tire load (corresponding with the tractor net weight) in Fig. 3.28 and the static axle load in Fig. 3.29. This principle allows the application to other tractor dimensions. It must be said that both diagrams result from Western European conditions – as guidelines. Additional measurements with prototypes are always recommended.

Regarding 4WD clutches for limiting the maximum torques (Fig. 3.29), two philosophies exist: with or without clutch. The author recommends torque limiting by a wet friction clutch if at all possible.

Application of Fig. 3.28. The rainflow history shows the number of cycles for 1000 h including all operations – in this case again being typical for Europe. The "mixture" of operations [3.25] should be secured or modified for specific projects. The cycle number must be adjusted to the expected tractor life, in the case of 10,000 h for example by a factor of 10.

It is recommended to start *FEM analysis* for the structure using the higher peaks – with adding high traction or braking forces. *Cumulative damage calculations* are recommended. They are easy to perform, if *S-N curves in the load scale* are available.

Load histories like Fig. 3.28 are also very useful for *accelerated design verifications by lab tests*. The lower loads are omitted (not contributing to damage), achieving considerable savings of time and test rig costs.

Application of Fig. 3.29. Load collectives with time portions are usually applied for life calculations of gear wheels and antifriction bearings [3.40]. *Rainflow load spectra are unsuitable in these cases!*

The first step is to transfer the sum frequency (Fig. 3.29) to a class frequency of torques. The linear partial damage can now be calculated for each class against an S-N curve in the torque scale (T-N curve). The sum of all the partial damage values is used to forecast the expected life (explained more in detail in chapter 5).

Efficiency of driven tractor axles.

Fig. 3.30 gives a full power survey for rear planetary axles with wet brakes [3.41], **Fig. 3.31** shows a complete efficiency map for a driven front axle [3.42]. The values are achieved by measuring the idling losses (at controlled temperature) and calculating the load-related additional losses.

Recommendations to reduce the idling losses (wet brakes!) see [3.41] and [3.42].

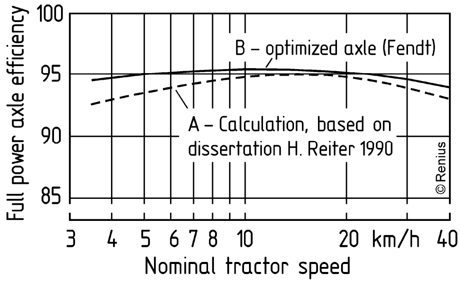


Fig. 3.30: Full power efficiency of a planetary rear axle with wet brakes. A - "good practice" [3.41], B - "fine-tuned design for efficiency"

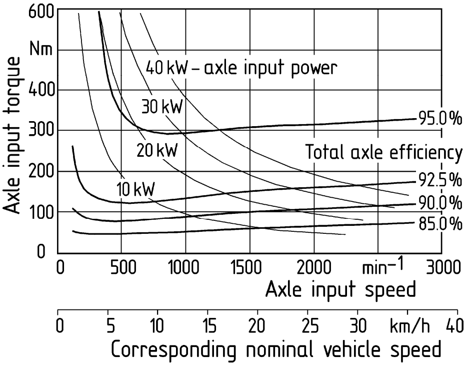


Fig. 3.31: Efficiency map of a front axle with wet disc brakes (Fig. 3.23). 90 kW tractor. Oil temperatures: differential 50 °C, planetary 60 °C [3.42]

3.7 Brakes

General requirements. Brakes are necessary to decelerate and to fix the tractor for on-road driving and for hillside operations. There are usually three groups of brakes:

1. *Service brakes*, **Table 3.5**
2. *Emergency brakes* (if service brakes fail)
3. *Parking brakes* (only for stop position)

No. 2 and 3 are often combined.

Table 3.5: General requirements on service brakes and legal requirements within the EU

- Relevant standards covered for
  - safety
  - public traffic
  - human relations
- One-side braking (turns)
- Controlled heat generation
  - within the brake
  - near the brake
- Protected from water and dust
- No vibrations and/or screetch
- Low drag losses
- Low level of wear
- Low maintenance
- Low initial and repair costs

Delegated regulation (EU) 2015/68 requires for maximum tractor mass:

Top speed:	≤ 30 km/h	> 30 km/h
Factor X of equation (3.3)	92	130
Average deceleration, m/s2	≥ 3.55 *	≥ 5.00 *
Max. pedal force, N	≤ 600	≤ 600
Max. hand force, N	≤ 400	≤ 400

* for the very right term of equation (3.3)

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Maximum stopping distance $s_{\max}$ is limited within the EU using the following equation:

$$s_{\max}\,[\mathrm{m}] = 0.15 \cdot V\,[\mathrm{km/h}] + \frac{V^2\,[\mathrm{km^2/h^2}]}{\mathrm{X}} \qquad (3.3)$$

V is the initial speed.
Factor X is given by Table 3.5 (EU).
Rating of performance see **Fig. 3.32**.

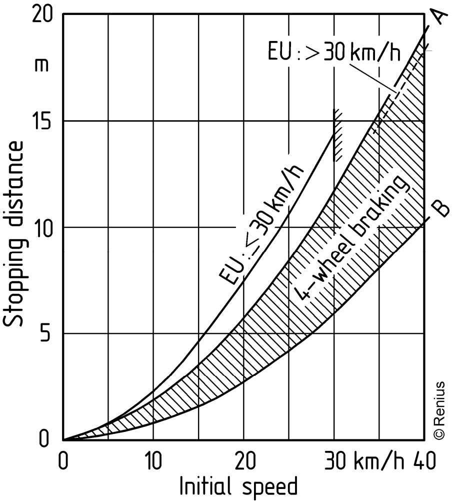
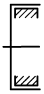
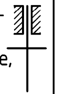
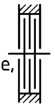
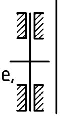


Fig. 3.32: Stopping distances and EU 2015/68. A - 4WD braking, brake with poor adjustment to axle load ratio front/rear, B - correct adjustment

Brakes only at the rear axle are cheap and often sufficient for a top speed of 30 km/h.

4-wheel brakes are needed in Europe for (popular) top speeds of 40 and 50 km/h. Top speeds above 60 km/h are subjected to stronger regulations for emergency cases. Typical tractor brake types see **Table 3.6**.

Table 3.6: Basic types of tractor brakes [3.19]

Concept	Pros	Cons
Drum brake, dry 	Lowest initial cost, easy to repair, no drag losses	High scattering of braking torques, sensitive to water, dust and mud (if not encapsulated)
Calliper brake, dry 	Able to handle very high sliding speeds, lining easy to replace, no drag losses	Wear from dust and mud, wear at lining and disc. High temperatures (neighboring parts)
Full disc brake, dry 	Longer life than calliper brakes, almost no drag losses	High initial costs, sensitive to mud (if not encapsulated). Lining often difficult to replace
Ring disc brake, wet 	Very low wear (lifetime possible). Resistant against water, dust and mud. Zero maintenance	Drag losses at higher speeds – needs special plate clearance, oil with adapted specifications, special know-how

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Practical brake design should be adjusted to technology levels, **Table 3.7**.

Drum brakes with mechanical actuation linkage are the classic low-cost concept.

They can offer a certain level of comfort if the inner amplification factor is high.

This is the case with servo brakes (friction force supports inner actuating force). But the disadvantage of high inner amplification factors is a *high sensitivity of the braking moment to different friction coefficients left and right,* which can hinder keeping the tractor on track or creating problems meeting regulations.

Wet disc brakes have become popular during recent decades for levels II–V.
They can be designed for lifespan and maintenance-free. Attention has to be paid to oil specifications, heat transport [3.43], and drag losses at higher speeds. A special mechanism for lifting the plates during the no-load position can reduce the shear stresses and related power losses [3.41].

Brake actuation can be done mostly by pedal forces, but needs hydraulic power assist for large tractors because of high masses and comfort issues. In Europe, tractors with 60 km/h top speed already use ABS systems (hydraulic or pneumatic).

One-sided braking should be possible for sharp turns – automatic re-connection is required now within the EU above 40 km/h. "Pull-in-turn" also supports sharp turns.

Solutions for all-wheel braking are explained in **Fig. 3.33**. Configuration A was popular in a first phase while B is still used because of excellent performance and low costs. Both create low initial costs.

Concept C is expensive and has higher shear losses but offers the highest level of flexibility – important for the future.

Table 3.7: Brake concepts by technology levels

Technology level	Comfort level	Type of service brakes	Actuation of the brakes
I	Very low	Drum brake	Simple mechanical linkage
II	Low	All types	Hydraulic pedal force transfer
III	Medium		
IV	High	Mainly ring disc brake, some full disc brakes	Hydraulic pedal force transfer, for large tractors power-assisted
V	Very high		

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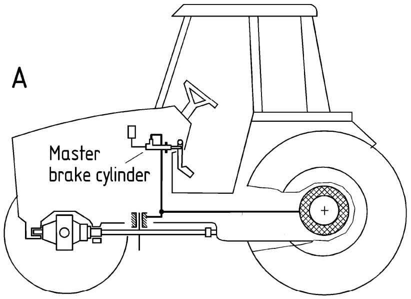
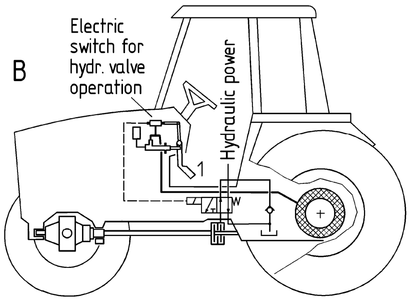
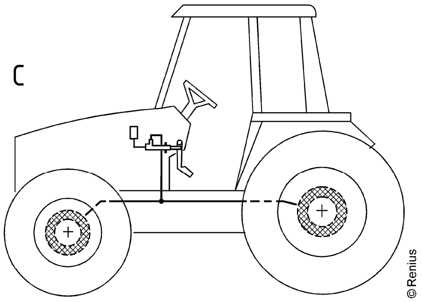


Rear brakes plus calliper brake at the 4WD shaft



Rear brakes plus automatically closed 4WD



Brakes for all wheels, usually between differential and planetary input – 2 circuits & ABS possible

Fig. 3.33: Concepts for all-wheel braking. In the EU, A and B require partial front differential locks. B offers excellent performance, C the highest flexibility. Costs increase from A to C

Integration of the brakes in rear axles. General customer demands are already mentioned above. *A general design rule is to put the brakes on shafts with high rotational speeds* in order to keep the torque and the brake size low – in favour of low costs, **Fig. 3.34**.

Concept A has been often realized, for example, with the Deutz transmission TW 50 (1966–81) in very high production numbers for low-cost tractor models. The drum brakes are directly connected with the driving wheels, while the emergency and parking brake is positioned on the faster countershaft of the final drive. The disadvantages of drum brakes (Table 3.6) can be reduced by dry calliper brakes.

Concept B was used by ZF decades ago. It has high rotational speeds for the brake, resulting in small dimensions and costs and is well protected from dust, mud, and water. It was, however, expensive to replace the linings and the dry brakes required proper reliable sealing on both sides.

Concept C is connected with a special final drive concept with two sets of spur gears. The position at the ends of the countershafts of the differential offers the possibility of using all types of brakes at high rotational speeds and with favourable access in the case of repair or replacement. This concept was popular in many tractors in the US and in Europe in the past but has been replaced widely by concept D.

Concept D can be called the modern standard configuration. It was mainly supported by the change from spur gears to planetary final drives (cheaper, higher reduction ratios, more compact). Wet disc brakes are common, often by only one plate with very large diameters – mostly located near the planetary ring gear.

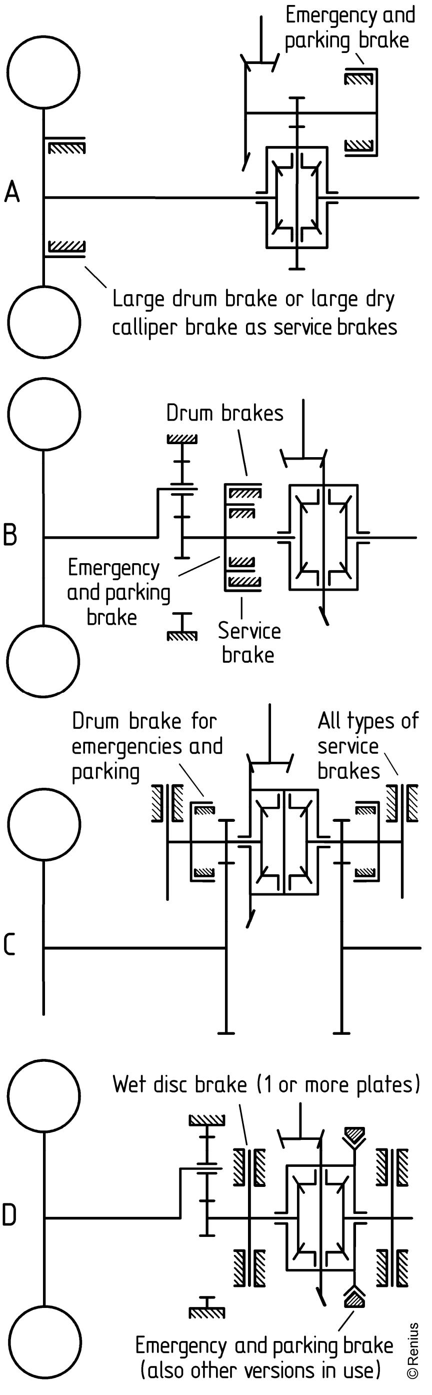

Emergency and parking brake
A
Large drum brake or large dry calliper brake as service brakes
Drum brakes
B
Emergency and parking brake
Service brake
Drum brake for emergencies and parking
All types of service brakes
C
Wet disc brake (1 or more plates)
D
Emergency and parking brake (also other versions in use)
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Fig. 3.34 (left): Four typical concepts of brake integration in tractor rear axles. Differential locks are used but not shown

Performance of the basic brake types (Table 3.6) can be characterized by the inner amplification factor C^* [3.44]:

$$C^* = \frac{\text{Circumferential braking force}}{\text{Inner brake loading force}} \qquad (3.4)$$

The higher this number, the lower the actuating force for a given braking force.

Disc brakes and drum brakes have different loading mechanisms, **Fig. 3.35** and thus also different amplification factors, **Fig. 3.36** (same notification 1, 2, 3, 4, 5).

Drum brake No. 5 has the highest amplification C^* in this survey, enabling a high comfort level. Its disadvantage is the very sensitive reaction to variations of friction coefficients. This type is therefore not recommended for service brakes but can be used well for emergency brakes.

Disc brakes have a low (linear) value C^* which is easy to calculate:

$$C^* = 2 \cdot \mu \cdot z \qquad (3.5)$$

where

μ = coefficient of friction

z = number of plates (each with 2 surfaces)

Factor 2 addresses friction on both sides. A dry calliper with a typical level of $\mu = 0.4$ has a value of $C^* = 0.8$ – straight line 1.

Wet disc brakes run in contact with transmission oil having very low friction coefficients of only about 0.06 to 0.16 depending on the layer material (high values for special organic materials and special oils, low values for bio-oils). The low friction values can be compensated for by a higher number of plates, a larger diameter and higher loading forces.

The effective braking radius r_m is

$$r_m = \frac{2}{3} \cdot \frac{(r_a^3 - r_i^3)}{(r_a^2 - r_i^2)} \qquad (3.6)$$

where

r_a = outer effective lining radius

r_i = inner effective lining radius

Drag torques have been investigated by *Reiter* [3.41].

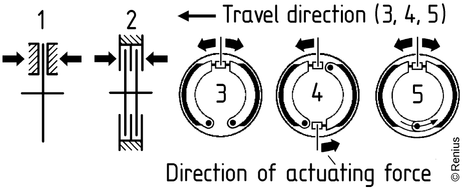

Fig. 3.35: Basic types of brakes; 1 calliper brake, 2 multi-disc brake, 3 simplex brake, 4 duplex brake, 5 duo-servo drum brake

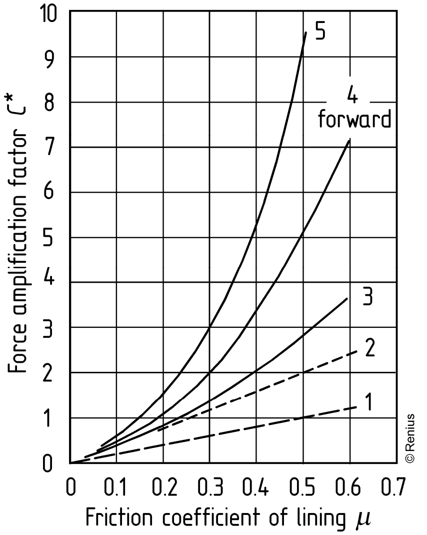

Fig. 3.36: Force amplification factor C^* for different brake types 1-5 as defined in Fig. 3.35. No. 4 has lower C^* when driven in reverse [3.44]. 3, 4, 5 can vary based on design details, 1, 2 can be increased by special ramps.

If oil fills the clearance between the plates completely and groove influences are neglected, the drag torque [3.41] is

$$T_{\text{shear}} = \frac{\pi}{2} \cdot \eta \cdot \Delta\omega \cdot \frac{z}{h} (r_a^4 - r_i^4) \qquad (3.7)$$

where r_a and r_i are as defined above and

η = dynamic oil viscosity

$\Delta\omega$ = difference in angular velocity

z = number of shear areas between plates

h = axial clearance between the plates

Low temperatures create high shear via η.

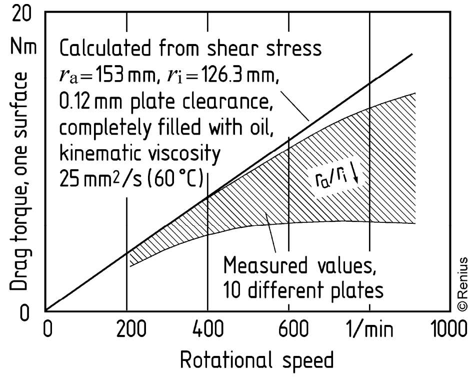

Fig. 3.37: Drag torque. Measured values for better comparison normalized on the geometry of the calculated plate, done by the ratio of the individual geometry factor $(r_a^4 - r_i^4)/4$ [3.41]

Based on equation (3.7), **power losses by shear** increase with $\Delta\omega^2$. However, in order to save energy and CO_2 emissions, the design of wet disc brakes calls for low idling losses, mainly driving on the road with high $\Delta\omega$ In order to limit the losses, clearance h should be at least 0.2 mm, e. g. forced by a retraction mechanism.

If so, measured values of the drag torque are below values for a completely filled clearance, **Fig. 3.37**. A high value of r_a / r_i is also in favour of low idling losses due to better conditions for disposing of the oil by centrifugal forces [3.41].

An additional rule recommends *not using plain plates* because of high adhesion.

Along with higher tractor speeds above 40 km/h, wet front axle brakes on the high-speed side of the planetary have become popular, **Fig. 3.38** (take note of the retraction by coil springs). This concept can also be used for **ABS systems** [3.45].

It is, however, not yet sure which system will finally be successful for tractors: ABS with hydrostatic brake operation (as used for passenger cars) or with pneumatic power (as used for trucks).

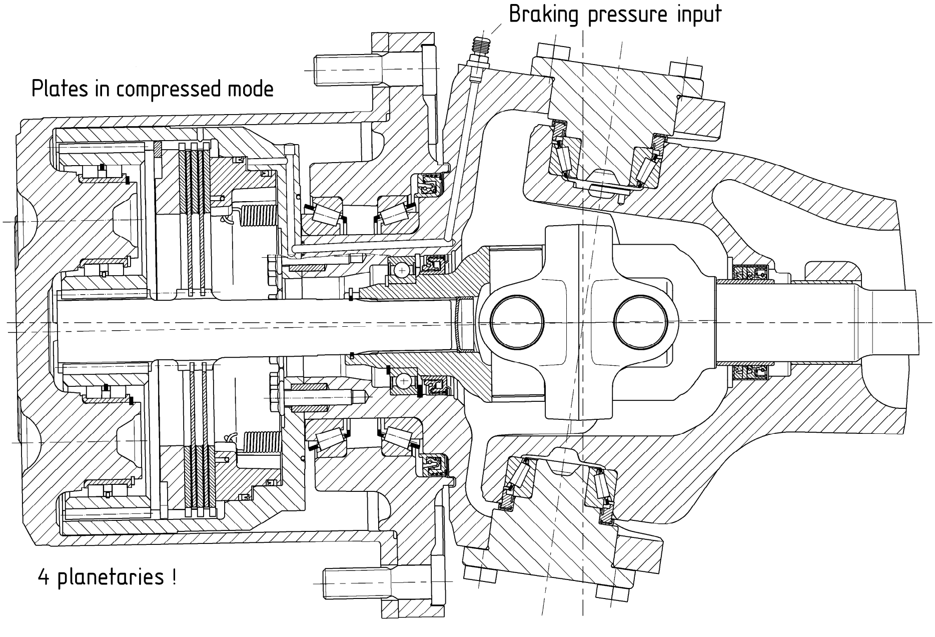

Fig. 3.38: Modern front axle head of the ZF “TerraSteer” tractor axle family. Courtesy ZF (2016)

3.8 Steering systems

3.8.1 Kinematics and forces

Some general requirements are listed in ISO 10998 including “steer by wire”.

“Ackermann” steering [3.46], **Fig. 3.39, drawing A** is the typical design for standard tractors. All four wheels should have a common static pole of motion to offer well-working motion in turns. The mathematical condition is:

$$\cot \delta_a - \cot \delta_i = \frac{b}{l} \tag{3.8}$$

If the small distance e is neglected, the track turning radius R_S can be expressed as

$$R_S \approx \sqrt{l^2 + \left(\frac{l}{\tan \delta_i} + s\right)^2} \tag{3.9}$$

The effective outside turning radius is larger by about half of the front tire width.

Concept B: All-wheel steering is seldom applied to standard tractors, but is interesting for special vehicles. It offers a lower turning radius. **All-wheel crabbing steering** is derived from B, for example, compensating for hillside drift.

Concept C: articulated steering is typical for narrow track tractors but also for very large tractors with equally-sized tires (often with duals or even triples). It offers multi-pass benefits in turns as well; however, it is not suitable for high speeds.

Concept D: Combined steering (CNH) of “articulated” and “Ackermann” results in a very low turning radius with reduced clearance problems [3.47].

Ackermann steering has conflicts with 4WD if there is no central differential, resulting in some disadvantages:

- *Front axle speed deficits in turns*
- *Circulating power on the road (wind-up)*

These two problems have led to automatic disengagements of the 4WD in these two cases – not just being ideal.

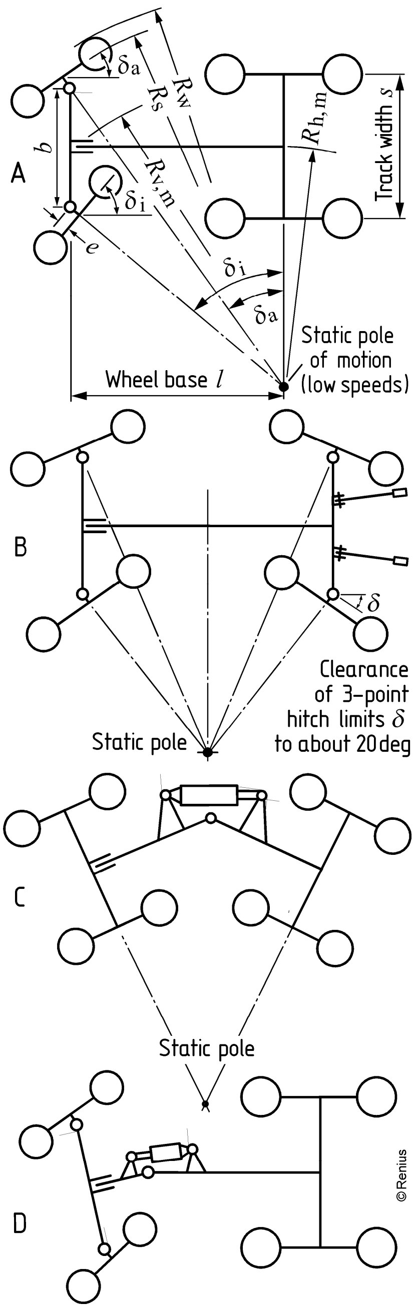
δa
Rw
Rs
Rh,m
Track width s
b
A
Rv,m
δi
e
δi
δa
Static pole of motion (low speeds)
Wheel base l
B
δ
Clearance of 3-point hitch limits δ to about 20deg
Static pole
C
Static pole
D
©Renius

Fig. 3.39: Kinematics of basic steering systems. Definitions see also ISO 10998

Steering kinematics in turns: An over speed demand of the front axle yields from the radii $R_{\mathrm{v,m}}$ and $R_{\mathrm{h,m}}$ (Fig. 3.39) [3.48]:

$$\frac{R_{\mathrm{v,m}}}{R_{\mathrm{h,m}}} = \frac{\dfrac{1}{\sin\delta_{\mathrm{i}}} + \dfrac{1}{\sin\delta_{\mathrm{a}}}}{2 \cdot \cot\delta_{\mathrm{i}} + \dfrac{b}{l}} \tag{3.10}$$

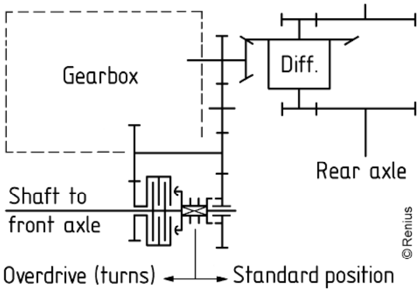

Fig. 3.40: Kubota “Double Speed Turn” front axle overdrive, introduced 1986 in Japan

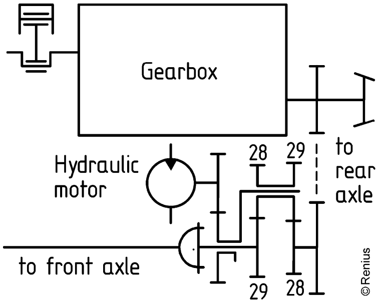

Fig. 3.41: Small planetary arrangement to control the 4WD power distribution front and rear. Research prototype at TU Munich 1995

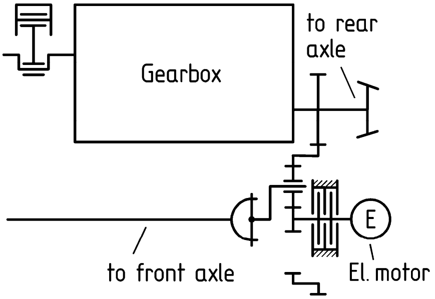

Fig. 3.42: Planetary arrangement to control the 4WD power distribution front and rear. Proposal of a John Deere study 2016 [3.52] and of ZF

A typical value for $\delta_{\mathrm{i}} = 55°$ is about 1.4, which means *that the speed input has to be increased for such a turn by 40%!*

High over speed demands are slightly reduced at higher speeds when the dynamic pole of the motion is moved forward as compared to the static pole (Fig. 3.39). Disconnecting the front drive automatically in turns is used but not ideal.

A Japanese design works with a one-step front axle overdrive in the case of sharp turns, first introduced by Kubota in 1986 [3.49], **Fig. 3.40**. This solution became very popular in Japan.

An infinitely variable pull-in-turn function was first offered for tractors in series from 1969 by the hydrostatic “front assist” drive of John Deere [3.50] with, however, limited success in Europe.

Research on the pull-in-turn and the wind-up problems was done at TUM under the author by *Grad* [3.51] and *Brenninger* [3.48] developing an additional planetary unit controlling the 4WD power distribution between front and rear axle, **Fig. 3.41**.

A small hydraulic motor drives the carrier of a double planetary with variable speed producing the right front axle rpm under all conditions – manly in sharp turns and on-road. The motor power was mostly very low, as its torque was only 7.4% of the total shaft torque and its speed only correcting. Highest power was required in sharp turns (high front axle over speed) and for front end loading (high torques). Control was done by steering angle, speeds, and torque distribution.

The system was integrated into a Fendt 312 LSA (85 kW) tractor with best results but came not in series production [3.38].

Another structure for such a “correcting” CVT, now with an electric motor, was part of a John Deere study [3.52], **Fig. 3.42**, similarly also proposed by ZF. The clutch can fix the sun gear for conventional mode.

A central differential was applied at the Schlüter Super 1500 V with ZF transmission T 350 in 1966 [3.53].

A similar structure is shown in **Fig. 3.43**. Torque distribution is fixed, but can be overridden by the locking clutch.

An interesting new hydrostatic solution was presented by AGCO-Fendt in 2015 with the new “VarioDrive” of the new large 1000 tractor line [3.54]. The front axle is driven by one of the two hydrostatic motors of the power split configuration. Locking of the central differential effect is possible, see chapter 5.

The **steering moment at the kingpin** is very high for *stationary steering on dry concrete,* increasing with steering angle to a maximum value according to the kinematics, **Fig. 3.44**, and the tire elasticity [3.55].

The kingpin moment is the sum of:

- the *lifting* moment T_W
- the max. *turning resistance* moment T_f

The lifting moment T_W is (from [3.56]):

$$T_W = F_W \cdot e \cdot \sin\varepsilon \cdot \cos\varepsilon \cdot \sin\delta \qquad (3.11)$$

with e and ε as defined in Fig. 3.44
and the steering angle δ in Fig. 3.39

The dominating part T_f can be calculated for dry concrete after *Wittren* [3.57]:

$$T_f = F_w \cdot f \cdot \sqrt{\frac{I}{A} + e^2} \qquad (3.12)$$

where

F_W = vertical tire load, f = from **Fig. 3.45**,
I = polar moment of inertia of tire print area,
A = tire print area, e = kingpin offset.

If the tire print is modelled as a circle with diameter b, the model is simplified to

$$T_f = F_w \cdot f \cdot \sqrt{\frac{b^2}{8} + e^2} \qquad (3.13)$$

A comparison of (3.12) with measurements [3.58] shows that moderate accuracy can be achieved in the case of realistic tire contact width (real load and pressure). Additional measurements are recommended – for example by metering the hydrostatic pressure of the power steering.

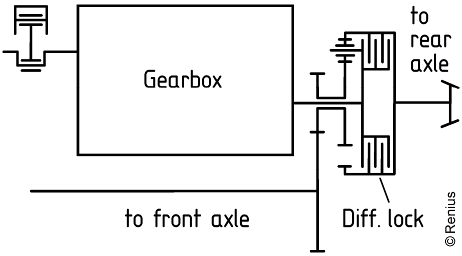

Fig. 3.43: Central differential for a torque portion of max. about 40% for the front axle

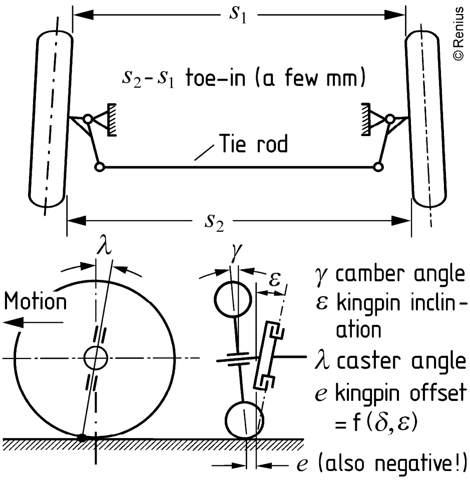

Fig. 3.44: Front wheel Ackermann geometry. Giudeline (4WD, standard track width): $\gamma < 2°$, $\varepsilon = 3...8°$, $\lambda = 0...3\ (12)°$, $e = -50\ ...\ +100$ mm

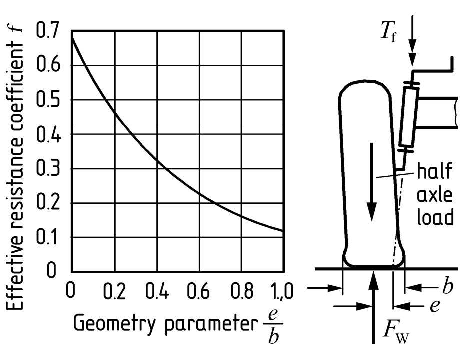

Fig. 3.45: Turning of air-inflated rubber tires on dry concrete for a stationary tractor. Lifting moment not included [3.57]

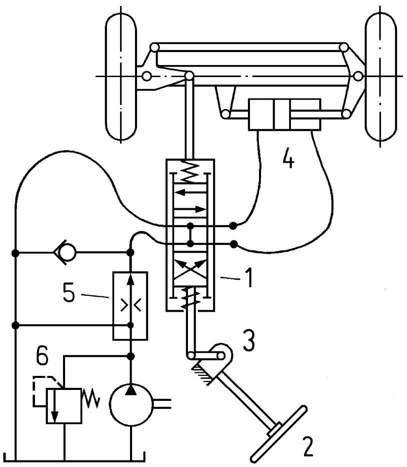

Fig. 3.46: Tractor power steering design **A:** Hydro-mechanical [3.31]. Symbols ISO 1219

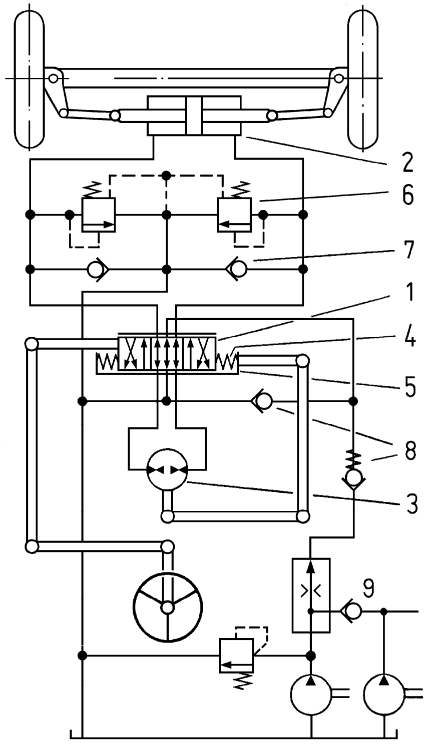

Fig. 3.47: Tractor power steering design **B** with purely hydrostatic actuating system [3.31]. Symbols from ISO 1219

3.8.2 Power steering

Power steering has been introduced along with increasing tractor power and weights for improved comfort with a special focus on front end loading and front implements.

Design principles can be classified by tractor technology levels, **Tables 3.8** and **3.9**.

Table 3.8: Actual power steering design principles and technology levels for standard tractors

Technology Levels	I	II	III	IV	V
Only manual	X	(X)			
A Hydro-mechanical		X			
B Hydrostatic		X	X	X	X
C Hydrostatic, automatic functions				X	X

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Table 3.9: Power steering designs with relation to on-board hydraulics for standard tractors

Techno-logy level	Comfort level	Power stee-ring concept	On-board hy-draulics
I	Very low	No power steering	one-pump open center systems
II	Low	Type A	Improved open center systems
III	Medium	Type B	
IV	High	Type B and C	Closed center load sensing systems or se-parate pumps
V	Very high		

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Concept A, Fig. 3.46, is a simple add-on with a steering drag link, which is "interrupted" by the steering valve (1). When the driver operates the steering wheel (2), a displacement at the valve (1) is created by the steering gear (3) which opens the line to the steering cylinder (4). Steering action is fed back to the valve case (1) so that the valve spool position comes back to its neutral position, finalizing the oil flow.

For every position of the steering wheel (2), this sequence control enables corresponding axle angles. The valve needs a constant flow, which is delivered by the flow control valve (5). The surplus flow bypasses the pressure relief valve (6).

Emergency steering is possible when the spool gets in contact with its case.

Concept B is the most-used for standard tractors with 4WD and technology levels III/IV/V. It is purely hydrostatic steering without any mechanical links, **Fig. 3.47**.

The control principle is, however, similar: The driver operates the valve (1) in order to open the line from the pump via the flow control valve to the steering cylinder (2). But (on its way) the flow must pass the metering unit (3), which moves the valve (1) back to its neutral position by the symbolized connecting linkage.

The outcome of the hydro-mechanical feedback is a *sequence control* as already mentioned for concept A. The valve (1) and the metering unit (3) are usually combined on one axis as the "steering unit".

Emergency steering is different to concept A: The inner part of the steering valve is moved against the springs (4), arrives at stoppers and moves the outer valve (5) in order to drive the metering unit (3) via the symbolized connection as a hand pump.

The pressure relieve valves (6) prevent extremely high pressure peaks coming from unusually high steering forces; for example, when contacting a wall. If they open, oil volume deficits can be compensated by the suction check valves (7).

The check valves (8) are needed in the case of emergency steering for oil suction control. Surplus oil flow of the flow control valve is not wasted in this case as it is used by joining at position (9) the on-board hydraulics and, for example, boosting the lifting speed of the front end loader.

Hydraulic on-board systems of technology levels IV and V usually work with load sensing (LS) systems, as popular for higher tractor power classes with variable displacement pumps. The steering valve is modified for LS mode, now with a constant pressure drop (automatic control by the pump displacement, additional LS pipe) [3.31]. A priority valve within the high pressure line offers adequate safety.

Concept C is based on concept B, but extended for automatic steering, **Fig. 3.48**. The add-on contains a proportional valve (1), two unlockable check valves (2), a pressure reducer (3) for low control pressure, and an electronic control unit (4). Inputs (5) are signals from sensors, from GPS, and from ISOBUS.

Articulated tractors need steering systems which require much higher performance:

The energy of an actuator work output for a lock-to-lock turn can be about 3.5 times higher than for Ackermann steering, assuming the same axle loads [3.57].

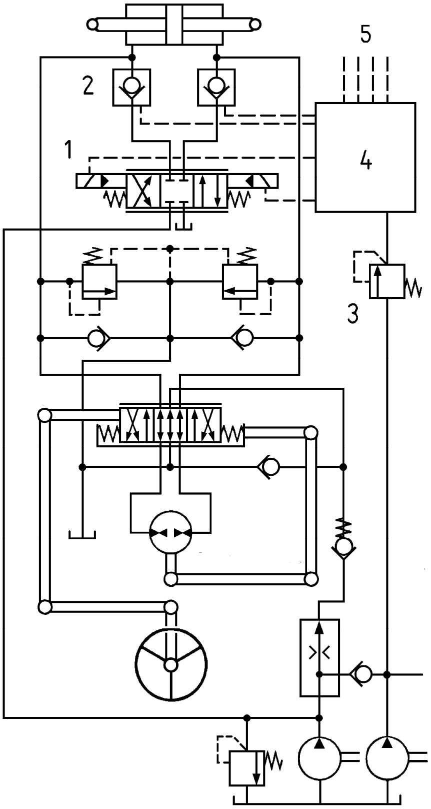

Fig. 3.48: Tractor power steering concept **C** offering automatic steering. The applied principle is based on [3.59]. Symbols ISO 1219

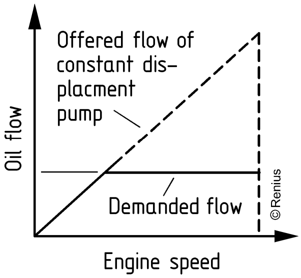

Fig. 3.49: Oil flow of a constant displacement pump versus engine speed

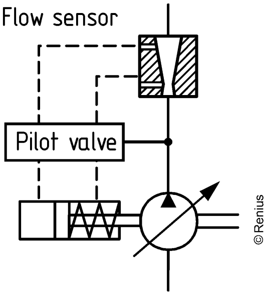

Fig. 3.50: Closed loop control of a constant oil flow for a power steering hydraulic circuit, proposed by the author in 1985 [3.60]

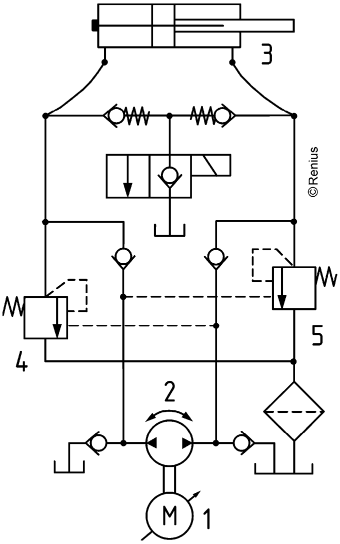

Fig. 3.51: Hydro-electric power steering for minimized energy consumption [3.31]

Reducing energy consumption of power steering. This has become important for all vehicles in order to improve fuel economy and to reduce the CO2 footprint.

The valves of common power steering systems (e. g. Pos. 1 in Fig. 3.47) need a sufficient constant oil flow even when the diesel engine is idling.

If the oil flow is supplied by a fixed displacement pump, this pump needs a constant flow valve in addition to the main steering valve, as demonstrated by the previous figures. For several decades, this was the standard solution for many vehicles. It is cheap, and hence still important for tractors of the lower technology levels.

The flow control valve delivers a constant flow independent of the engine. As the pump must be able to deliver the needed oil flow even for an idling engine, the displacement must be relatively high. This results in an important flow surplus at higher speeds, meaning working speeds, **Fig. 3.49**. A high flow rotates in the system even during "non-steering" phases creating a lot of wasted energy.

One possibility of reducing the losses is to use the surplus flow for the on-board hydraulics, as shown in Fig. 3.47 and 3.49.

A next step could be to combine the steering hydraulics with the load-sensing on-board system. Even better is a pump with an adjustable displacement and automatic flow control, **Fig. 3.50**, proposed by the author [3.60] and realized years ago for large Schlüter tractors,.

As the average net steering power is extremely low, a hydro-electrical solution was introduced 2007 by Jungheinrich for forklift trucks, **Fig. 3.51**. The electric motor (1, variable speed) drives the constant displacement pump (2) which works on the left or right side of the steering cylinder (actuator 3). The valves (4) and (5) enable the flow back from the low-pressure side of the steering cylinder.

Steering systems for fully-tracked tractors. The first generations were usually equipped with conventional differential units and clutches and brakes on both sides of the final drive in order to brake left for a left turn and right for a right turn while the outer side final drive supports the turn by a certain overspeeding. This could be supported by simultaneously disconnecting a clutch on the braking side.

Such a simple system is no longer accepted for modern systems of higher technology levels, as the steering behaviour is not very comfortable and – in addition – different in response when driving down steep slopes.

This was the reason for developing systems which replace the classical differential by an active control of the speed output left and right by planetary units with superimposed speeds.

The basic principle is shown in **Fig. 3.52**. There is no differential, only a simple bevel gearing. The ring gears of the final drive planetaries are, however, not fixed, but their rotational speed is controlled by a small hydrostatic motor and a bevel gearing unit in both directions. The hydraulic motor is driven by a variable oil flow generated by a variable displacement pump. The main advantage of such a system is that the average vehicle speed is not influenced in turns or when driving downhill.

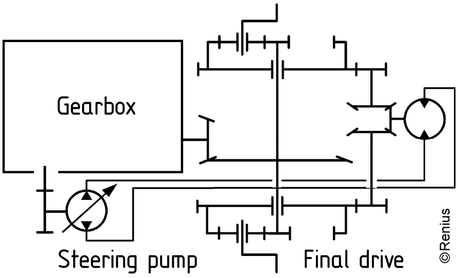

Fig. 3.52: Active hydrostatic differential steering system for the drive axle of a tracked vehicle

This principle was applied to tanks – but also to the first agricultural rubber-tracked tractor from Caterpillar, **Fig. 3.53**. The bevel gearing was replaced by a system of spur gears, also preventing any influence on the vehicle speed when turning.

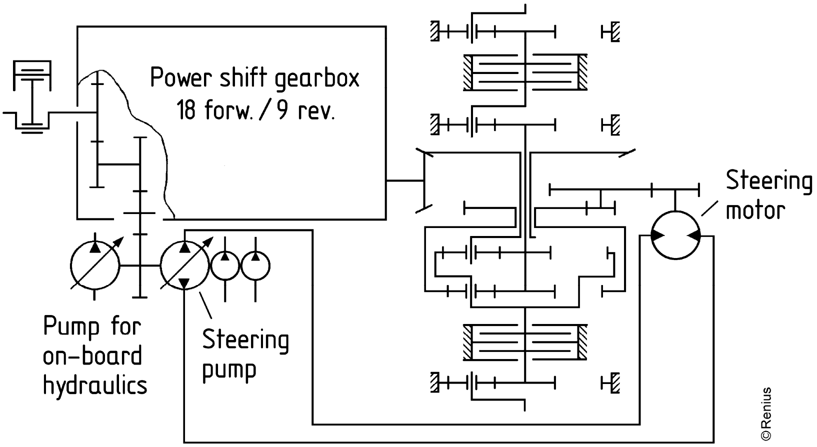

Fig. 3.53: Differential steering of the rubber-tracked Challenger CH45 1987 (nominal power 178 kW, net weight 10t) [3.61]. A similar system was applied for the Fendt 900Vario MT (2018)

3.9 Track width and wheel fixing

An **adjustable track width** enables proper row crop operations; examples for given row spacings are listed in **Table 3.10**.

Table 3.10: Recommended track width and tire width for some row crops and their typical row spacings (distance from row to row center)

Type of row crop	row spacing, cm	Max. tire width in.	cm	Possible track, mm
Potatoes, etc.	62.5 75	9.5 12.4	24 31	1250, 1900 1500, 2200
Corn (maize)	60 75	15.5 16.9	39 43	1800 1500, 1600 2200
	75 80	18.4 20.8	47 53	1500, 2200 1600, 2400
Sugar beets, Vegetables	45 50	9.5 9.5	24 24	1350, 1800 1500, 2000
Wheat	12.5,16.6	9.5-18.4	24-47	several[1]

[1] permanent lanes kept free of plants

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There are several typical designs of track width adjustments:

- Standard rims to be turned: two tracks
- Rims with brackets, resulting in eight tracks, **Fig. 3.54**
- Special front axles, **Fig. 3.55**
- Infinitely adjustable tracks, see the following page, **Fig. 3.57**

ISO standard 4004 recommends the track widths **1500 – 1800 – 2000 mm**, each with ± 25 mm tolerance.

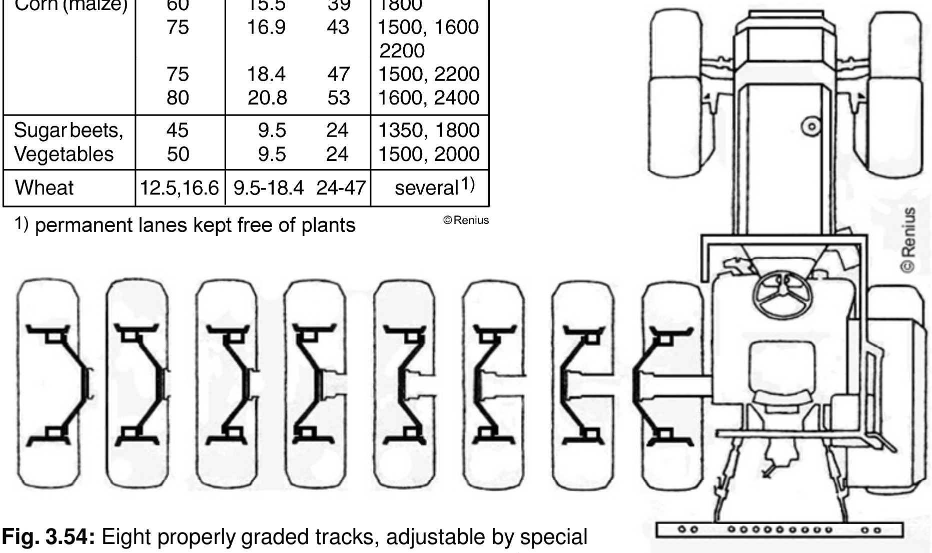

Fig. 3.54: Eight properly graded tracks, adjustable by special bolted rims which can also be turned. Typical range of tracks: From 1524 mm (60 in.) to about 2200 mm (about 90 in.)

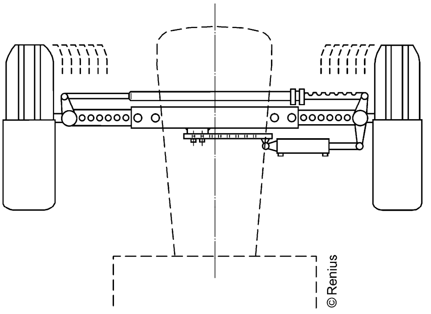

Fig. 3.55: Track adjustment of a non-driven front axle (tractor with rear-wheel drive)

Many farmers do not like to change the track width of their tractor(s) and/or other machinery often.

That's why they like to use a fixed system of tracks for as many operations as possible, working for a long time with one constant track spacing of their tractors, equal with farm machinery and trailers.

But the variety is necessary to face all row distances and track width practices which are used worldwide.

Wheel-to-hub fixing is standardized in ISO 5711. Some important measures are listed in **Fig. 3.56** and **Table 3.11**.

A flexible wheel fixing for stub axles enables an infinitely variable track adjustment (tractor axle lifted), **Fig. 3.57**.

Controlled traffic [3.62–3.65] reduces soil compaction or even prevents it, **Fig. 3.58**.

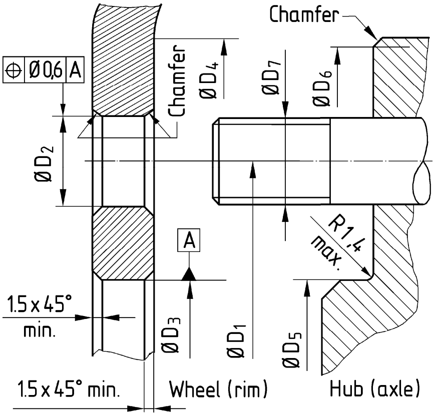

Fig. 3.56: Flat attachment type of wheel-to-hub fixing, measures recommended by ISO 5711

Table 3.11: Dimensions of wheel-to-hub fixing in mm as standardized by ISO 5711

Equally spaced stud holes	Wheel				Stud	Hub	
	D_1 nom	D_2 $^{+1}_{0}$	D_3 $^{+0.5}_{0}$	D_4 min	D_7 1)	D_5 $^{0}_{-0.2}$	D_6 $^{0}_{-5}$
4	100	15	61	145	12	60.8	140
5	140	17	96	185	14	95.8	180
6	205	21	161	255	18	160.8	250
8	203.2	21	152.4	257	18	152.2	252
	275	24	221	325	20	220.8	320
10	335	26	281	390	22	280.8	385
12	425	26	371	470	22	370.8	465

1) For information only All dimensions in mm

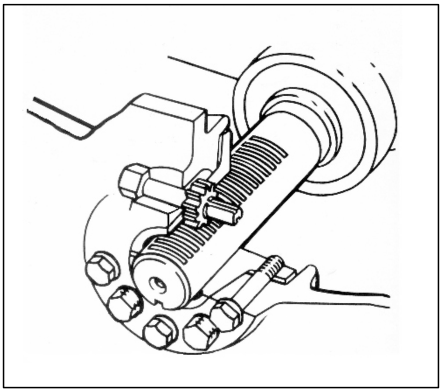

Fig. 3.57 (right): Infinitely variable track width of stub axles by a “rack and pinion” system. Courtesy John Deere

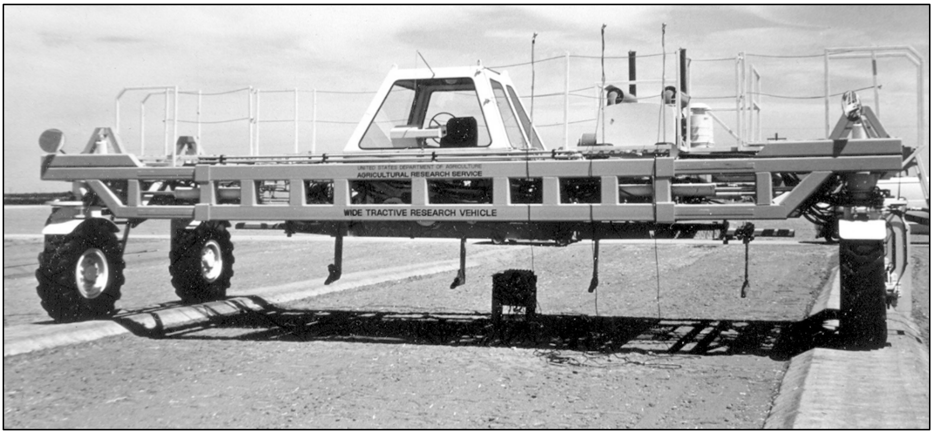
UNITED STATES DEPARTMENT OF AGRICULTURE
AGRICULTURAL RESEARCH SERVICE
WIDE TRACTIVE RESEARCH VEHICLE

Fig. 3.58: Wide-frame vehicle developed for controlled traffic research by the United States Department of Agriculture, track width 10 m, concrete lanes [3.63]. Courtesy NSDL, Photo P10, 347a

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4 Diesel engines

4.1 Basic function and history

The name "diesel engine" recognizes its inventor *Rudolf Diesel* (1858–1913) [4.1].

His first patent DRP 67 207 was in power from Feb. 28, 1892 and documented on Feb. 23, 1893 by the German Imperial Patent Office Berlin. He presented a first running and working prototype in 1897.

Rudolf Diesel received an honorary doctorate from the "Technische Hochschule München" (today TUM) in 1907 [4.2].

The principle of a four-cycle diesel engine is as follows, **Fig. 4.1**:

- Stroke 1: Suction or charging of pure air
- Stroke 2: Air compression, fuel injection
- Stroke 3: Expansion, working stroke
- Stroke 4: Exhaust

This means that ignition is not done by spark plugs but by fuel injection into the compressed air, which is very hot due to the high compression ratio of about 1:16 to 1:19. This high ratio was initially *Diesel's* aim, to increase the thermal efficiency.

He had in mind realizing a thermodynamic cycle like **Fig. 4.2** with fuel injection timing just to get a constant pressure during the burning process.

Compression and expansion (Fig. 4.2) can be approximated by adiabatic changes:

$$p \cdot V^{\kappa} = \text{constant} \tag{4.1}$$

where

p = actual cylinder pressure
V = actual volume
κ = average isentropic exponent, about 1.35

The thermal efficiency of such an ideal cycle can be approximated by introducing the compression ratio $\varepsilon = V_1 / V_2$:

$$\eta_{th} = 1 - \varepsilon^{1-\kappa} \tag{4.2}$$

This means that the thermal efficiency increases considerably with the compression ratio – one reason for the low specific fuel consumption of diesel engines.

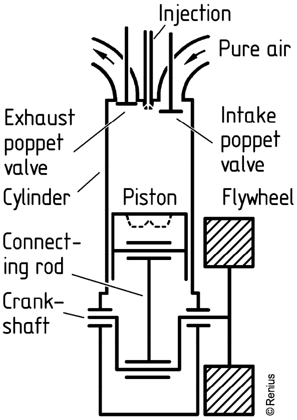

Fig. 4.1: Basic concept of a diesel engine. Ignition by injecting fuel under very high pressure into the highly compressed pure hot air

The real world diesel cycle can usually be understood as a combination of constant pressure and constant volume processes (during injection) also being not exactly adiabatic [4.3] due to heat losses. The final heat and pressure losses at 4 – 1 can be reduced by turbocharging.

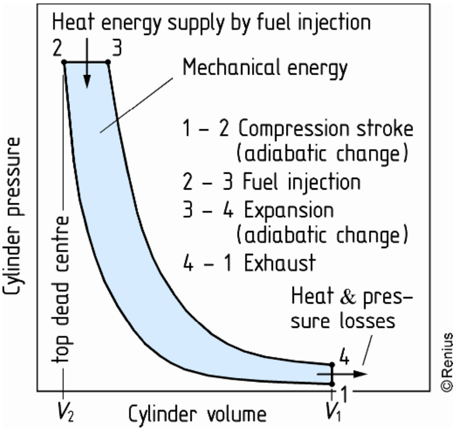

Fig. 4.2: Ideal working cycle of a diesel engine, simplified (air intake stroke neglected). No use of exhaust energy (4 – 1)

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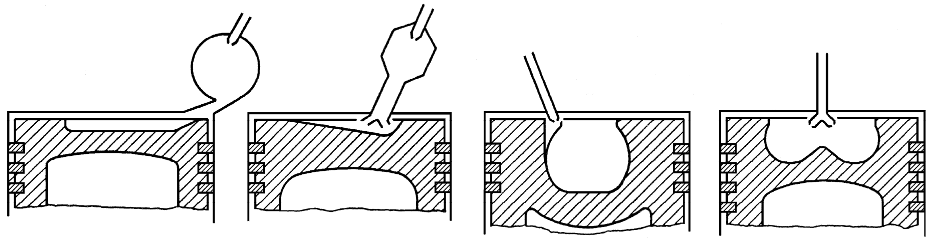

Turbulence chamber
1953 Borgward & Fiat
from 1976 used by VW

Precombustion chamber
1923 Benz-Sendling tractor
1936 Mercedes car (260 D)

Direct injection
MAN concept, one license
used for GDR tractor ZT 300

Direct injection
Common design, in
many modified versions

Fig. 4.3: Configurations of diesel engine burning chambers: Turbulence chamber, prechamber, and direct injection versions. Direct injection now clearly dominates

Diesel engines were initially not a standard power source for tractors (chapter 1.3). The first mass-produced tractor generations had ignition-fired Otto engines using mostly gasoline for warming up and the cheaper petroleum (“kerosine”) for the working time, see the Fordson F, the John Deere Waterloo Boy, and many other tractors.

The new age of diesel engines for vehicles started with tractors and trucks (not with construction machinery or passenger cars):

Benz-Sendling presented prototypes in 1922 and a series production of 100 units in 1923, using a precombustion chamber (MWM, *Prosper l’Orange*), **Fig. 4.3**. Its nominal fuel consumption was reported to be only 212 g/hph (288 g/kWh) [4.4].

MAN followed in 1924 with the first direct injection engine, first used for trucks [4.5].

The status of precombustion chamber engines was published in 1926 (by Deutz) [4.6], the status of the injection technology around 1930 by *Prosper l’Orange* [4.7].

A general replacement of ignition-fired engines by diesel engines took place for tractors and trucks in Europe from about 1930 – mainly driven by fuel economy (fuel was more expensive than in the US).

The first Mercedes diesel car 260 D, presented in 1936, opened the diesel era to cars, using the pre-chamber design.

After World War II, diesel engines established themselves as a general standard for tractors in Europe. In most cases, the first generation still had pre-chamber heads although direct injection was already known [4.8] – even by *R. Diesel* [4.5].

It was mainly its better fuel economy driving this change – in Europe for tractors and trucks from about 1955 [4.9, 4.10], for cars later (Fiat 1988, Audi 1989) [4.5].

An important further step was the introduction of turbochargers [4.11] supplying air with a certain over-pressure (about 0.8 to 3 bar) instead of “natural aspiration”.

This led to a higher power density and better service for working in high altitudes, which was one reason to develop charging technologies, for example, early for aircraft piston engines.

The first tractor with a turbocharger was the Allis-Chalmers D 19 (US), introduced in 1961. Many famers were sceptical in the first phase (durability?) but were finally convinced by the advantages. Other companies followed; for example, John Deere with the 4520 in 1969 [4.12].

Further engine developments achieved higher power density (turbocharging with high pressures and intercoolers), electronically controlled injection systems, higher injection pressures, longer service life, and reduced service and emissions.

4.2 Global market demands

Tractor specifications have increased considerably in bandwidth of functions and engine power over the last decades as already outlined in chapter 1.4 for the whole tractor. A uniform **"world" diesel engine** would be too expensive for regions like India, Africa, Asia, or South America. However, some **general global engine requirements** can be listed:

- Life: B10 (90% probability) of 6000 to 12000 h, increasing with the power level
- Low specific fuel and oil consumption
- Torque back-up at least 15%
- Ability for block or frame tractor chassis
- Easy to service and repair
- Easy to replace and to get spare parts
- Ability to be produced under license

Typical local conditions can be:

- Level of regulations
- Structure and level of agriculture
- Level of subsidies
- Level of labour costs
- Level of education & infrastructure
- Level of fuel quality & prices

Resulting specifications are summarized by five technology levels in **Table 4.1**.

The strong global competition in producing diesel engines requires global design approaches in order to achieve cost effective platforms for as many technology levels as possible with local customizing.

Customizing mainly addresses:

- Level of brake mean effective pressure
- Torque back-up
- Nominal rotational speed and its control
- Cylinder head (material, 2, 3, or 4 valves)
- Piston design and piston cooling
- Automatic valve play control or not
- Air cleaning system
- Air charging system, charging pressure
- Intercooling concept (if used)
- Fuel injection system
- 4-cylinder engine: Lanchester drive?
- Exhaust gas aftertreatment
- Cooling concept (ambient temperature?)
- Energy saving auxiliary drives
- Use of electrics and electronics
- Cab heating and air conditioning
- Service intervals
- Options

The specific initial costs of a diesel engine (per kW) of technology level I are about half of those of level IV or V. Production of low-level engines in highly developed countries is mostly not economic.

Table 4.1: Global classification of basic diesel engine specifications and specific costs by five technology levels (compare with table 1.4). Based on [4.13], updated

	Cooling		Charging*				Injection			Valving			Aftertreatment				Brake mean eff. pressure				Cost level (per kW)			
Technology level	Air	Water	Natural asp.	TC	TC + IC	TC+ IC+ VTG	Mech.<500 bar	Mech.>500 bar	El. > 1500 bar	2 Valves/cyl.	3 Valves/cyl.	4 Valves/cyl.	Nothing	EGR + DOC	EGR+DOC+DPF	DOC+DPF+SCR	Low	Moderate	High	Very high	Low	Moderate	High	Very high
I	X	X	X				X			X			X				X				X			
II	(X)	X	X	(X)				X		X			X	(X)				X			X	(X)		
III		X	(X)	X	X			X	(X)	X	X	(X)		X	(X)	(X)			X	(X)		X	X	
IV		X			X	X			X		(X)	X			X	X			(X)	X			X	X
V		X			X	X			X			X			X	X			(X)	X			X	X

* TC Turbocharging, IC Intercooler, VTG Variable turbine geometry

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4.3 Engine performance, fuel economy, and turbocharging

Table 4.2: Diesel engines for tractors: typical specifications. Power "without cooling fan"

Cylinder displacement	l	0.8 – 1.1 – 1.5
Hub to bore ratio	-	1.15 – 1.30
Rated engine speed	rpm	1700 – 2500
Rated power per cyl.	ISO-kW	12–15 (20–40)
Torque rise, ISO	%	15 – 35
Lowest specific fuel cons.	g/kWh	200 – 230*
Average oil consumption	g/kWh	0.2 – 0.3
Compression ratio	-	17–19 (16–18)
Ignition peak cyl. pressure	bar	70-85 (120-200)
Nominal mean eff. pressure	bar	6 – 7.5 (10 – 22)
Maximum injection pressure	bar	400* – 2500
Max. exhaust temperature	°C	500 – 600

( ) Turbocharged * Lowest technology level, low power

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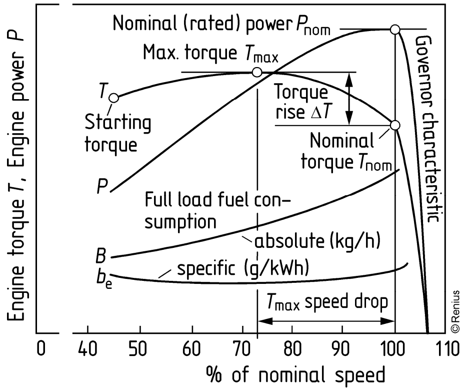

Fig. 4.4: Full load performance and fuel consumption of a tractor diesel engine

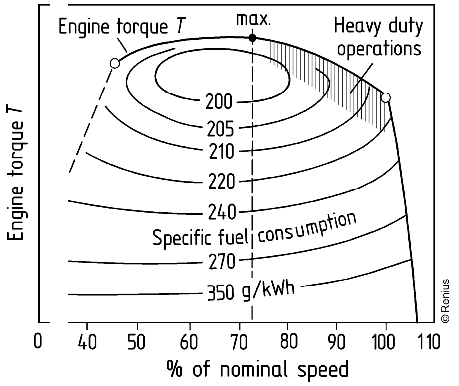

Fig. 4.5: Typical fuel efficiency map of a very efficient diesel engine (model). Definition of the power basis: "without cooling fan drive"

The power output of a diesel engine can be expressed by its output torque T and its angular output speed ω:

$$P = T \cdot \omega \tag{4.3}$$

Or with practical units:

$$P\,[\mathrm{kW}] = T\,[\mathrm{Nm}]\,\frac{\pi \cdot n\,[\mathrm{rpm}]}{30{,}000} \tag{4.4}$$

The *brake mean effective pressure* p_{me} is the average pressure during the working cycle representing the engine power density:

$$p_{me}\,[\mathrm{bar}] = 1200 \cdot \frac{P\,[\mathrm{kW}]}{V_{DC}\,[\mathrm{l}] \cdot n\,[\mathrm{rpm}]} \tag{4.5}$$

where

P = engine output power
V_{DC} = total engine cylinder displacement
n = engine rotational speed
1200 = constant value for 4-cycle engines

For typical specifications see **Table 4.2**.

Fig. 4.4 shows diesel engine characteristics at full load: performance and fuel consumption versus engine speed. Some large engines offer maximum power even slightly below the nominal speed.

The full load torque at the lowest speed should be above the nominal torque to allow easily starting up the tractor when pulling. This is no problem for natural aspiration but a challenge for turbocharged engines. A torque rise ΔT (also "torque back-up" or "torque reserve") is required in order to prevent a service brakedown in case of a torque demand above the nominal value. The torque rise is usually defined as

$$\Delta T\,[\%] = \frac{T_{max} - T_{nom}}{T_{nom}} \cdot 100 \tag{4.6}$$

The related speed drop should be about 25–30% for moderate torque rise, slightly higher in the case of high torque rise values (constant power engines).

Fig. 4.5 addresses the typical shape of the fuel economy of a very efficient modern diesel engine.

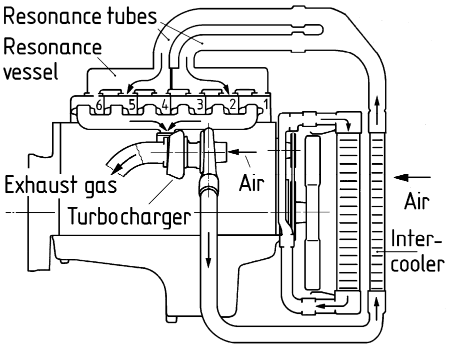

Fig. 4.6: Early adaption of a turbocharger with intercooler and resonance system for a 6-cylinder diesel engine, MAN D 2566 MK (11.4 l, 235 kW, 1979). Courtesy MAN

Modern diesel engines of the upper technology levels are mostly equipped with turbochargers, **Fig. 4.6** and **Fig. 4.7**.

The exhaust gas drives a turbine which is directly connected to an air compressor. Extremely high rotational speeds are necessary; for example, 120,000 rpm.

A high oil flow is needed to center the shaft and to cool the journal bearings. Air intercoolers reduce the temperature increase at additional pressures of above 0.8–1.0 bar.

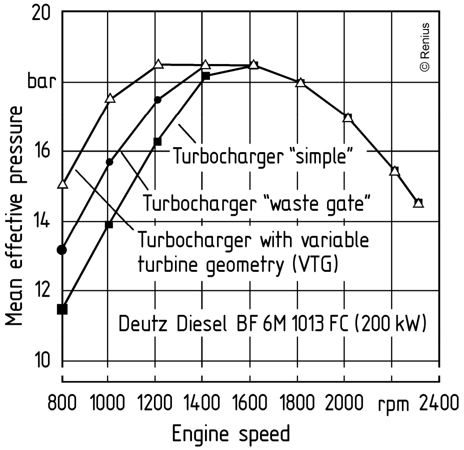

Fig. 4.8: Mean effective pressure of three turbocharger designs, full load. Data courtesy Deutz

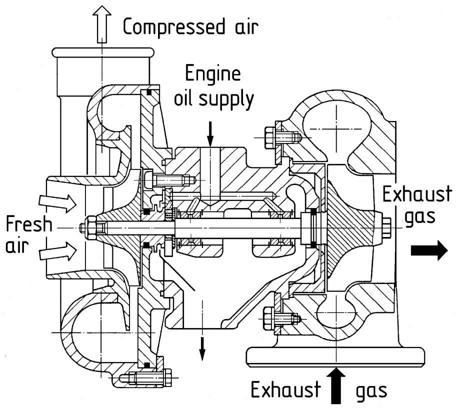

Fig. 4.7: Typical turbocharger design for a diesel engine. Courtesy KKK, modified

As turbocharger pressures increase progressively with their speeds, this can result in poor engine torques at low speeds.

Two strategies are common for a certain compensation, **Fig. 4.8** [4.14]:

- Oversize the charger power and bypass the surplus at high rpms by a *waste gate.*
- Use a charger with *variable turbine geometry* (VTG).

A VTG offers the best torque and fuel economy characteristics, **Fig. 4.9** [4.14].
A double charger can use further potential.

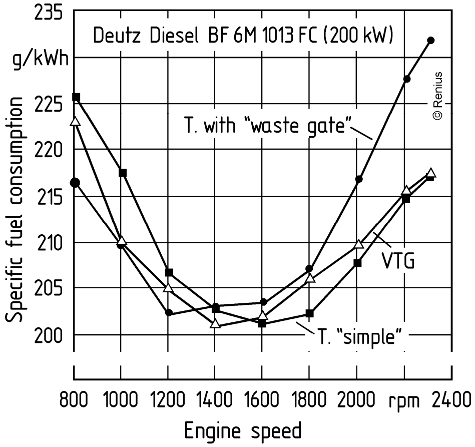

Fig. 4.9: Fuel economy of three turbocharger designs at full load. Data courtesy Deutz

4.4 Examples of mass-produced tractor diesel engines

The first example is a 6-cylinder turbocharged Deutz diesel engine with electronically controlled common rail injection, **Fig. 4.10**.

Its specifications are listed in **Table 4.3** on the following page. The high pressure pumps are operated by a special camshaft. The high specific power needs a very solid design with large diameters of the journal bearings at the crankshaft and the piston (compare with Fig. 4.12 and 4.13).

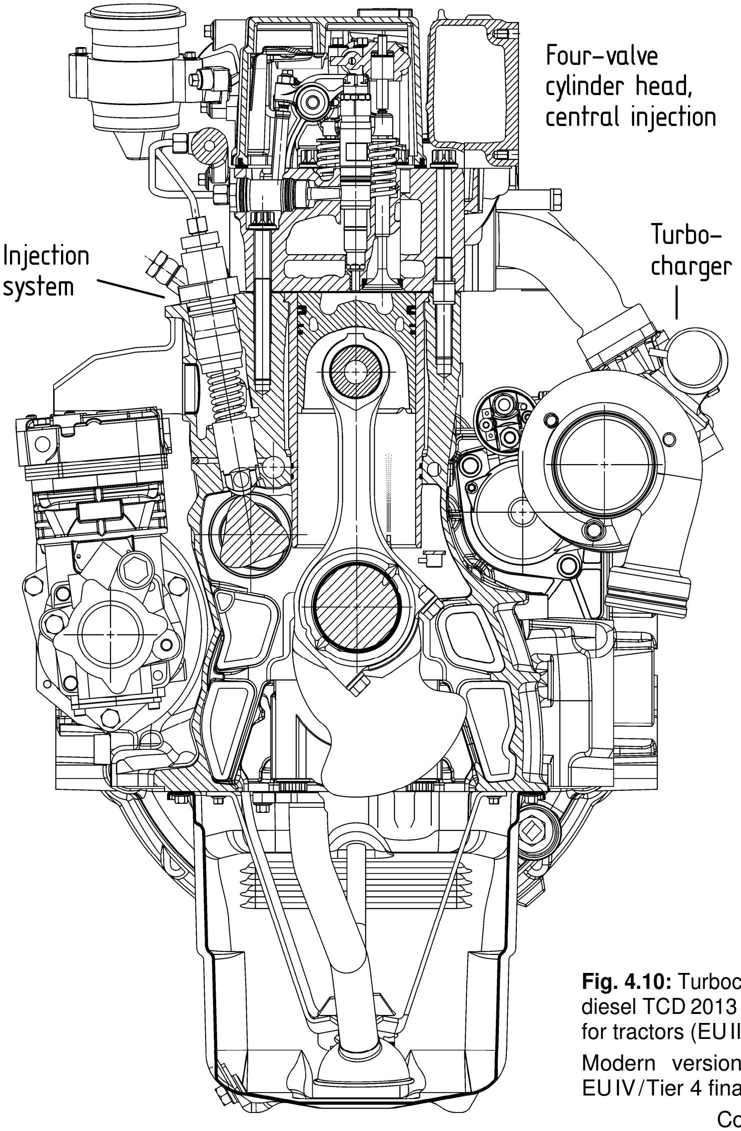

Fig. 4.10: Turbocharged Deutz diesel TCD 2013 L06 4V, used for tractors (EU III A/Tier 3).

Modern versions can meet EU IV/Tier 4 final and EU V

Courtesy Deutz

Fig. 4.11 illustrates the thermodynamic process for the compression and working cycle: The turbocharger creates an initial intake cylinder pressure of 3.06 bar. This is 4.06 bar absolute, hence about 4 times higher than with natural aspiration which explains the high power density. The air intake pressure is increased to about 110 bar by the piston compression and raised to 160 bar by the ignition due to the fuel injection into the hot air beginning at 11 deg ahead of the upper dead center finishing 6 deg after it. These 17 degrees correspond to only about 1.3 milliseconds.

The cylinder pressure decreases during the working cycle, until the exhaust valves open at about 110 deg crankshaft angle (about 70 deg ahead of the bottom dead center). The pressure is still high enough to drive the turbocharger which at the same time reduces the exhaust temperature due to the gas expansion.

Table 4.3: Specifications of the Deutz diesel TCD 2013 L06 4V (EU IIIA). Courtesy Deutz

Number of cylinders	-	6
Bore and stroke	mm	108/130
Displacement	l	7.15
Compression ratio	-	18.1 : 1
Valves per cylinder	-	4
Automatic valve play control	-	no
Nominal injection pressure	bar	1400
Turbocharger with waste gate	-	yes
Intercooler for turbocharged air	-	yes
Rated power ISO 14396*	kW	261
Rated speed	rpm	2200
Rated torque*	Nm	1133
Maximum torque*	Nm	1496
Speed at maximum torque	rpm	1450
Torque rise (back-up)*	%	32
Rated mean effect. pressure*	bar	19.9
Lowest fuel consumption*	g/kWh	195
Versions for Tier 4 final & EU V	-	yes

*without cooling fan

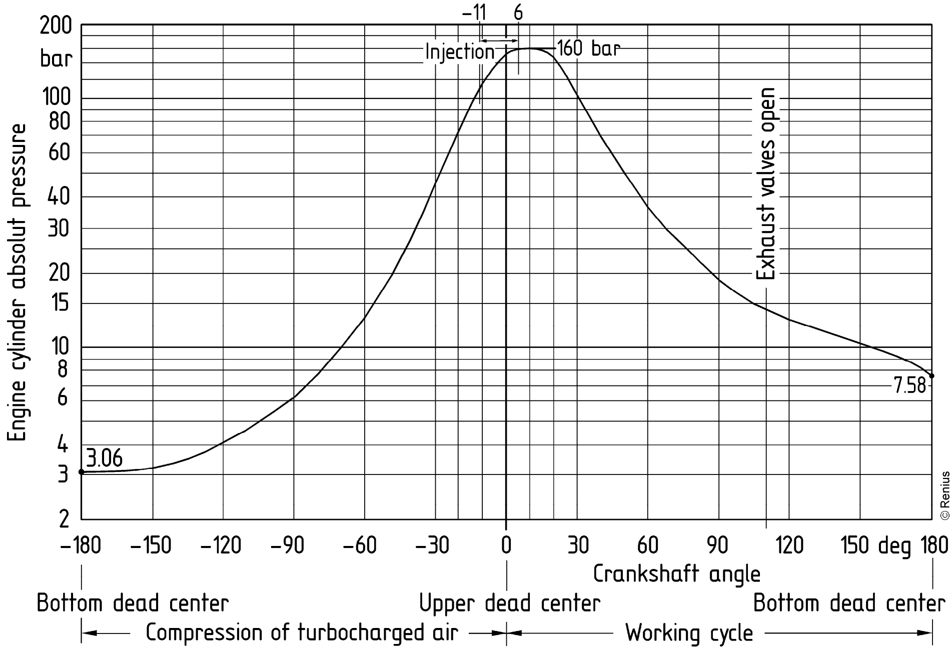

Fig. 4.11: Measured cylinder pressure for the diesel engine Deutz TCD 2013 L06 4V (Fig. 4.10) for 238 kW at 2200 rpm (EU III A). Data courtesy Deutz

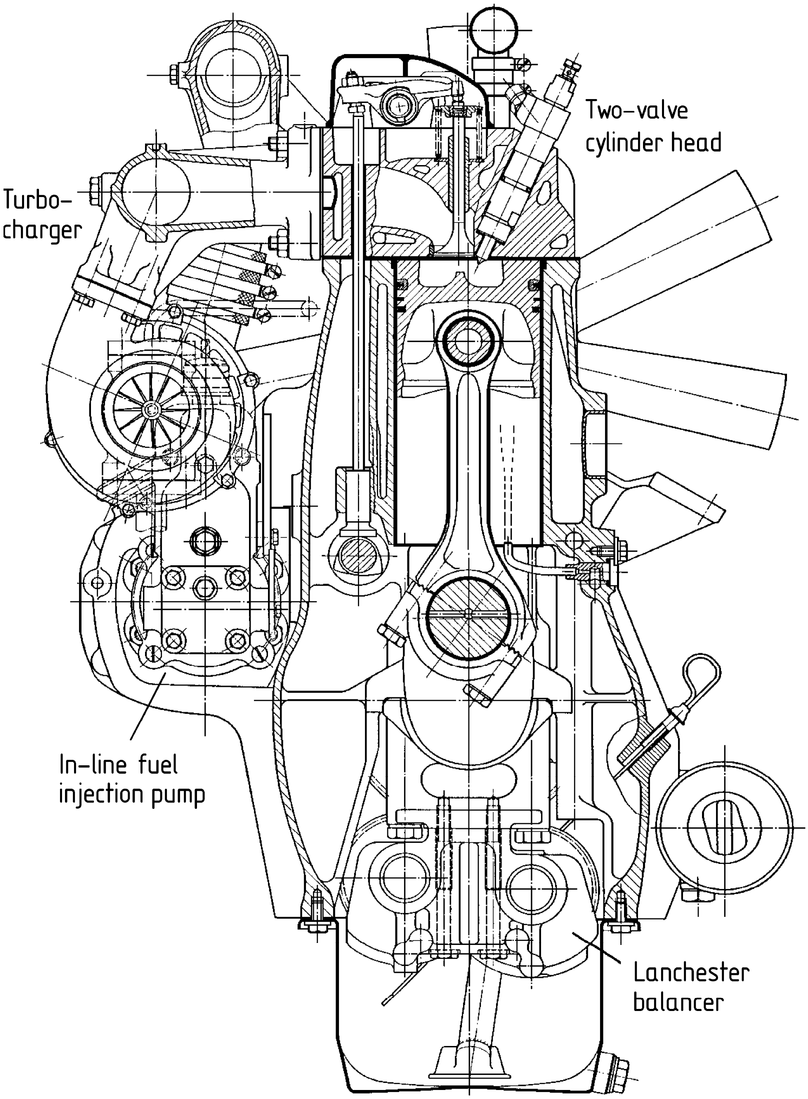

Fig. 4.12: 4-cylinder diesel engine MWM TD 226-4 with Lanchester balancer and moderate turbocharging pressure. Bore 105 mm, stroke 120 mm, displacement 4.16 l. Rated power up to 70 kW at 2350 rpm. Put into production around 1990 and, for example, applied to Fendt tractors. Lanchester balancers (Fig. 4.19) are often used for tractor block chassis designs in order to reduce chassis vibrations Courtesy MWM

Former diesel engine designs from highly developed countries can help lower-developed countries to find design solutions for their demands on lower technologies.

An interesting example is the MWM engine TD 226-4, **Fig. 4.12**, which was installed in the Fendt 300 tractor series (and others) for many years [4.15].

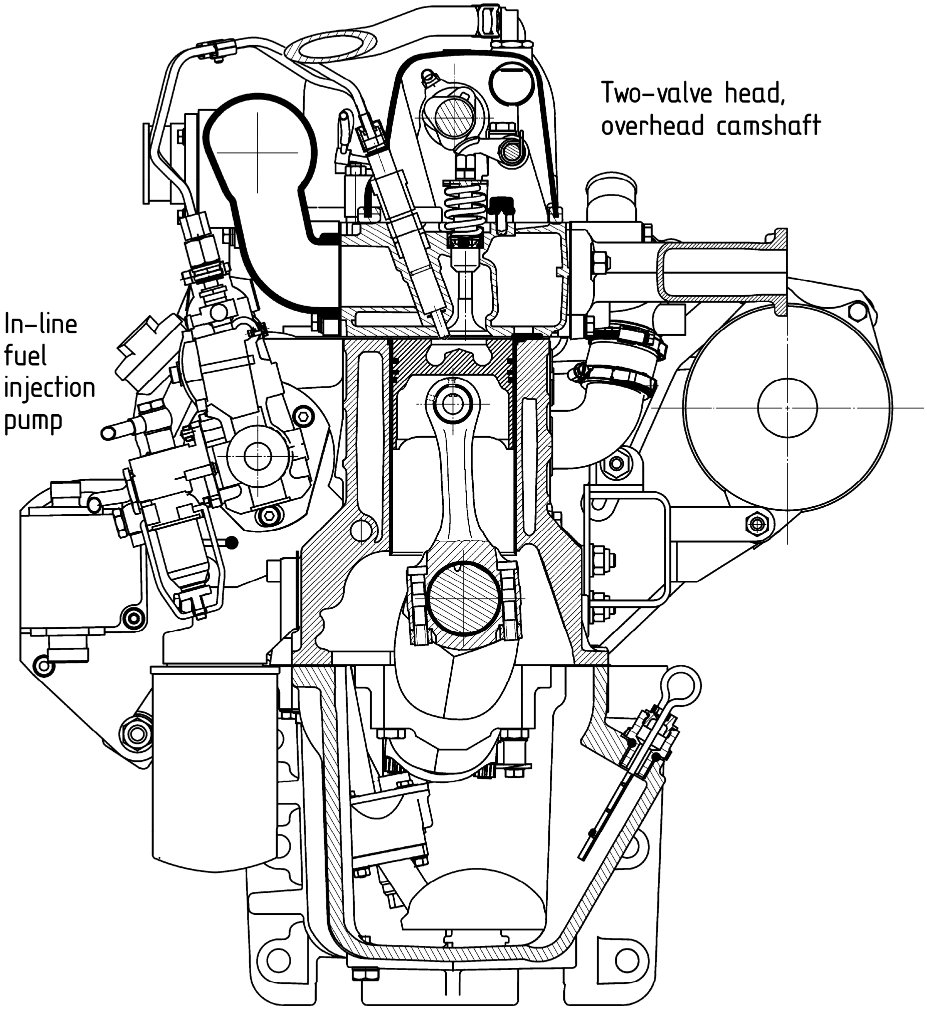

Fig. 4.13: 4-cylinder 2.6 l diesel engine of FORCE Motors, India, derived from the Mercedes-Benz car diesel engine OM 616. Bore & stroke 90.9 & 100 mm. Nominal power 33 kW (45 hp) at 2200 rpm. Mechanical inline pump, injection pressure 580 bar. Courtesy FORCE Motors Ltd.

The diesel engine of **Fig. 4.13** is a successful example of a technology transfer from Europe to India [4.16]. The Daimler-Benz car diesel engine OM 616 (1973–95) served as a basis. The main design changes with consultancy by AVL List, Austria are: Direct injection (replacing pre-chamber), increasing the stroke from 92.4 to 100 mm, reinforcing the block and designing a new cast oil pan (both for a robust block chassis), reinforcing the crankshaft, and some other modifications – all around 1997.

Typical (welcomed) relicts are the overhead camshaft and the plated rocker arms.

4.5 Diesel injection systems

Mechanical systems with about 300 to 500 bar injection pressure and inline or axial piston pumps have been the standard design for several decades. They are still in use for low technology levels, **Fig. 4.14**.

Injection valves of these systems are opened by the pressure, **Fig. 4.15**, needing a precise displacement control of the pump, **Fig. 4.16**.

The mechanism for the variable injection rate of inline pumps from Bosch is demonstrated by the movement of one of its pump pistons, Fig. 4.16.

The injection rate is controlled mechanically by indexing the piston by a rack and pinion system.

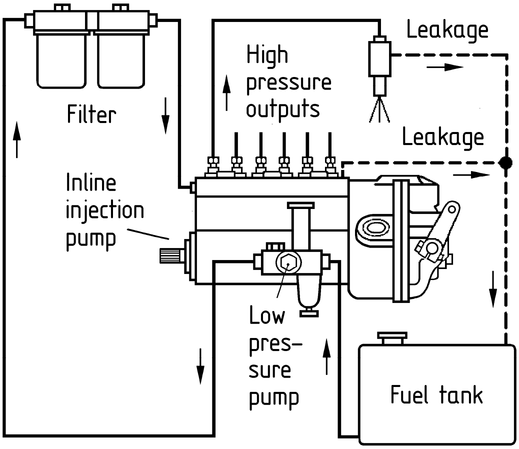

Fig. 4.14: Mechanical diesel injection system with inline piston pump. Courtesy Bosch

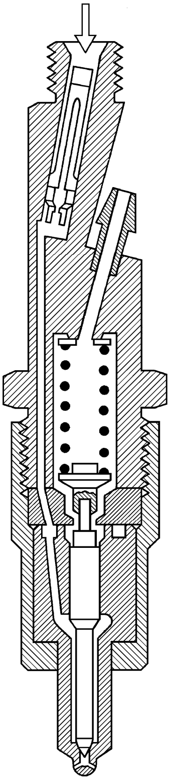

Fig. 4.15: Mechanical injection unit, which opens by pressure Courtesy Bosch

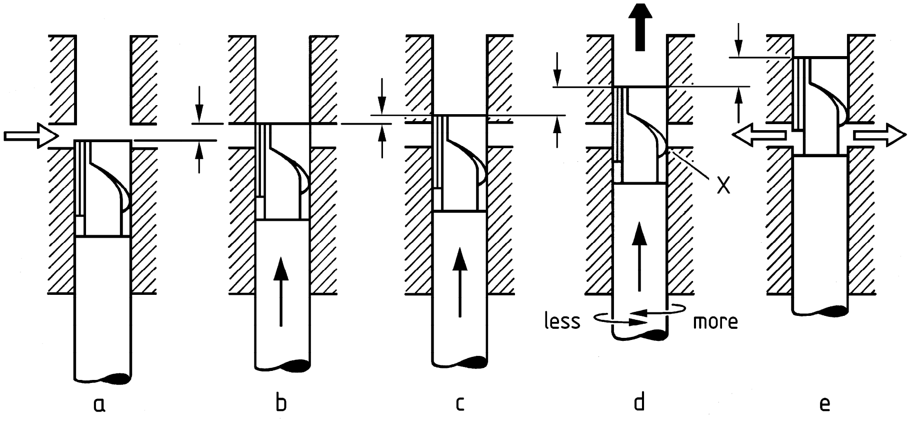

Fig. 4.16: Inline pump: Movement of one piston. a fuel inlet at lowest point, b closing of fuel inlet, c precompression, d movement for injection, limited by control edge X (rotational piston position is given by gear rack position), e movement without load. Courtesy Bosch

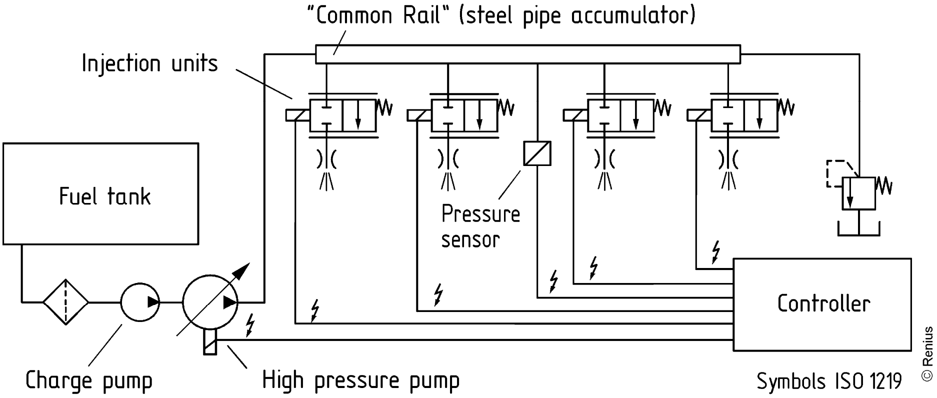

Fig. 4.17: Structure of a "Common rail" injection system with permanent high rail pressure [4.17]

Electronically controlled systems with injection pressures work with various types of piston pumps. "Common rail" systems are usual for the higher technology levels. Permanent nominal pressures are used up to about 2500 bar (2019), **Fig. 4.17**.

The fuel is compressed to a reduced volume due to the very high pressure level offering a certain storing effect like an accumulator, damping the flow discontinuity of the injections. Storing is also supported by the elastic widening of the pipes and the rail.

Injection units for common rail systems mostly work with electric solenoids producing magnetic pilot forces against springs, **Fig. 4.18**. In the case of no current (left), the control orifice is closed by the force of the upper spring and the ball type poppet. The common rail pressure is pressing down the piston-needle unit.

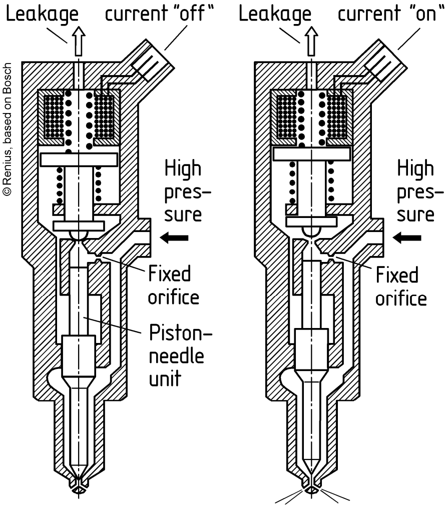

Fig. 4.18: Injection unit for common rail systems: Needle closed, when the current is "off" (left). Needle opens by a high current (right). Based on graphics of Bosch [4.18]

When the solenoid is activated, the magnetic force opens the ball type poppet and allows a certain flow through both orifices to the tank. This flow creates a pressure drop at the fixed orifice, resulting in a reduced pressure above the piston-needle unit, which is lifted up by the high pressure and opens the needle for injection (right). The whole injection procedure is completed in about one millisecond. The dynamics can be improved even more by an electric "piezo" actuator instead of the solenoid although its extremely small travel signals require a very high travel amplification. Piezo actuators are, however, currently not applied for tractor diesel engines (2019).

4.6 Noise, vibrations, and emissions

Engine noise is generated by mechanical contacts and by steep cylinder pressure rise gradients.

Engine noise always increases with the rotational engine speed, one reason to limit the speed. The following model was used for air-cooled Deutz engines [4.19]:

$$\Delta L\ [\mathrm{dB(A)}] \approx 40 \cdot \log\,(n_2/n_1) \qquad (4.7)$$

where

ΔL - noise level increase
n_1 - base of rotational speed
n_2 - increased rotational speed

Example: n_2/n_1 =2500 rpm/2050 rpm: $\Delta L \approx 3.5$ dB(A).

Vibrations. 4-cylinder in-line engines are compensated regarding the first order, as two pistons always move in the opposite direction [4.3]. The common center of gravity of all piston masses is, however, not stable for a complete rotation. It moves down a little in the intermediate positions between the dead centers due to the geometry of the crank mechanism.

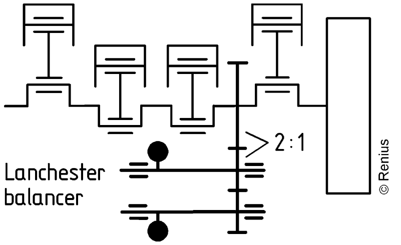

Fig. 4.19: Drive scheme of a Lanchester balancer as used for 4-cylinder diesel engines

The vibrations can be reduced by a Lanchester balancer, **Fig. 4.19**, (after *F. W. Lanchester,* UK, 1868–1946).

The two imbalanced shafts are driven at exactly double the engine speed, rotating in opposite directions. Such a system for 4-cylinder diesel engines is often applied for premium tractors along with block chassis designs.

Diesel engine emissions became a major issue when the US government enacted Clean Air legislation for the automotive industry (1970s), truck industry (1980s), and non-road industry in the 1990s [4.20, 4.21].

EU limits are now about the same for trucks and mobile machinery but slightly stricter than for passenger cars [4.22-4.24]. Criteria for the exhaust gas emissions are established for nitric oxide (NOx), hydrocarbon (HC), carbon monoxide (CO), and particulate matter (PM), **Table 4.4**. An EU stage V limits the number of particles (PN) and enlarges the addressed power spectrum below 19 and above 560 kW.

EU regulations considerably favour engines with rated power below 56 kW.

Table 4.4: Emission limits for diesel engines of mobile machines (EU regulations [4.22-4.24])

Nom. engine power, kW	Recent stage within EU					Stage V within EU				
	Stage EU	CO g/kWh	HC g/kWh	NOx g/kWh	PM g/kWh	CO g/kWh	HC g/kWh	NOx g/kWh	PM g/kWh	PN 1/kWh
0 < P < 8	-	-	-	-	-	8.0	7.5		0.4	-
8 ≤ P < 19	-	-	-	-	-	6.0	7.5		0.4	-
19 ≤ P < 37	IIIA	5.5	7.5		0.6	5.0	4.7		0.015	$1\mathrm{x}10^{12}$
37 ≤ P < 56	IIIB	5.0	4.7		0.025	5.0	4.7		0.015	$1\mathrm{x}10^{12}$
56 ≤ P < 130	IV	5.0	0.19	0.4	0.025	5.0	0.19	0.4	0.015	$1\mathrm{x}10^{12}$
75 ≤ P < 130	IV	5.0	0.19	0.4	0.025	5.0	0.19	0.4	0.015	$1\mathrm{x}10^{12}$
130 ≤ P < 560	IV	3.5	0.19	0.4	0.025	3.5	0.19	0.4	0.015	$1\mathrm{x}10^{12}$
≥ 560	-	-	-	-	-	3.5	0.19	3.5	0.045	-

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For tractors, the **load level** of the standardized **load cycle** is above practical values (mainly regarding small tractors); however, the *test definition in g for performed kWh* widely compensates for the load level.

Exhaust emissions for tractors (within the group of mobile machinery) have been reduced dramatically in the EU (and similarly in the US), **Fig. 4.20**, in only 15 years.

The reduction of the two most important pollutants – NOx and particulates – has been above 95% (stage IV). Tremendous human and financial investments had been undertaken to achieve this result.

New aftertreatment components were developed along with turbocharging, electronic injection control, and injection pressures up to 2,500 bar, **Fig 4.21**. The treatment accounts for about double the total engine initial costs as compared to stage I units. Main components are:

1. EGR – **E**xhaust **G**as **R**ecirculation
2. DOC – **D**iesel **O**xy **C**at
3. DPF – **D**iesel **P**articulate **F**ilter
4. SCR – **S**elective **C**atalytic **R**eduction

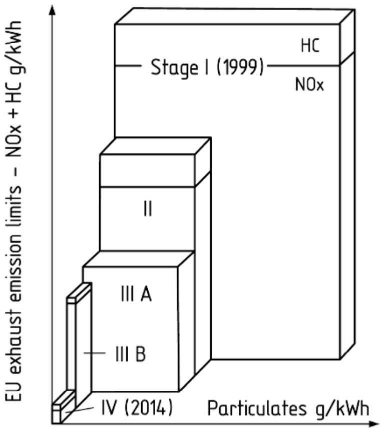


Fig. 4.20: Development of EU exhaust emission limits for mobile machinery > 56 kW © VDMA

The NOx reduction with SCR can be expressed (simplified) by the formula:

$$2\,NH_3 + NO + NO_2 \rightarrow 2\,N_2 + 3\,H_2O \qquad (4.8)$$

AdBlue consists of 32.5% urea in water for processing NH_3 on-board. Optimized modern tractor engines need only a few % urea in relation to the diesel consumption.

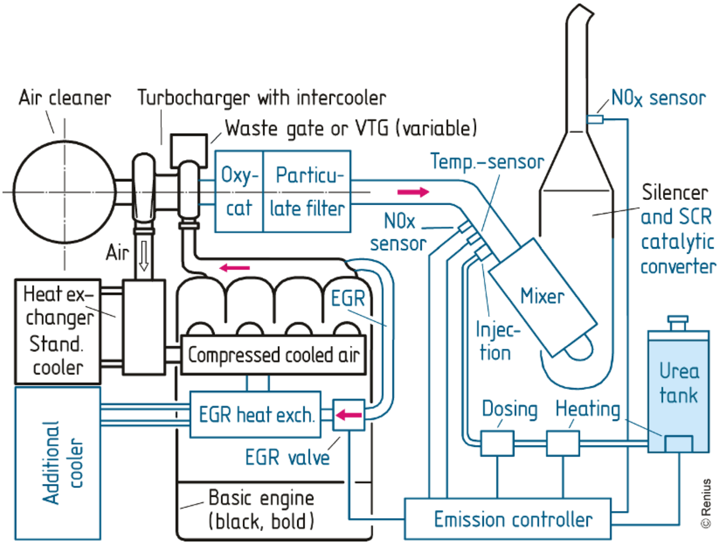


Fig. 4.21: 4-cylinder diesel engine with typical exhaust gas aftertreatment components (blue) meeting EU stage IV (US Tier 4 final) and EU stage V. Black bold lines address the basic engine

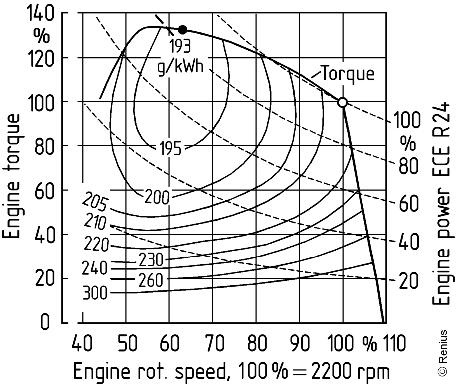


Fig. 4.22: OECD fuel consumption map of the Fendt Favorit 822 diesel engine, MAN 6-cyl., 154 kW at 2,200 rpm, ECE R 24, 6,87 l, turbocharged

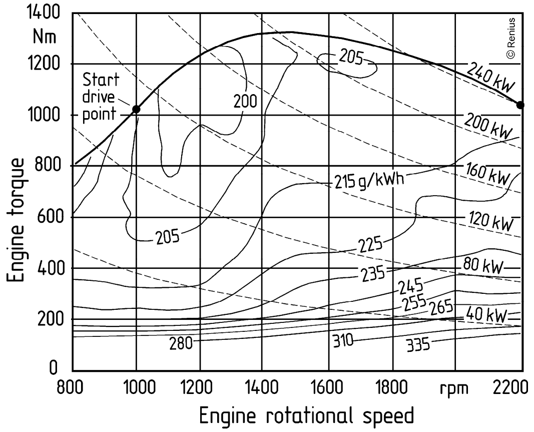
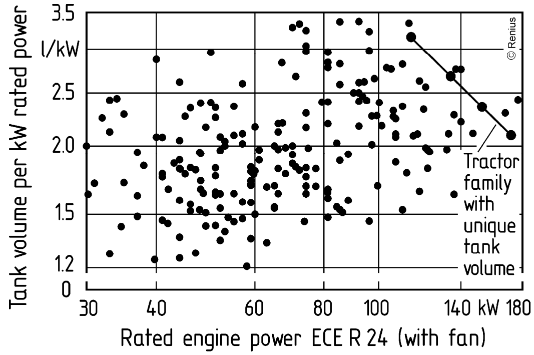


Fig. 4.23: Fuel consumption of Deutz diesel engine TCD 2013 L06 4V (without fan) Courtesy Deutz



4.7 Fuel for diesel engines

4.7.1 Consumption, tank size

An **adequate tank volume** is a function of three parameters:

- Fuel economy of the engine
- Tractor duty cycle
- Working time until refuelling

The **fuel economy** of a diesel engine can be evaluated best by maps representing the **S**pecific **F**uel **C**onsumption (SFC) in g/kWh.

Fig. 4.22 reproduces values from OECD Test 1532 (Fendt 822, 1994). This level is outstanding, but represents a period of only moderate emission regultations.

A second map is plotted in **Fig. 4.23** for the Deutz engine of Fig. 4.10. Injection timing is configured in order to achieve a large area of low fuel consumption addressing partial loads.

Engine fuel consumption maps have been modelled with satisfying results for example in [4.25].

Specific tank volumes were analyzed in 1990, **Fig. 4.24** [4.26]. They show an increasing trend versus rated power as larger tractors are often used more intensively (chapter 1, [1.5]).

Current specific fuel consumptions of diesel engines and actual duty cycles of tractors are very similar to those of 1990 which still allows using the survey diagram of Fig. 4.24 as a guideline.

The log-log net has the advantage that individual points of a tractor family with a common tank size are represented on a straight line.

Fig. 4.24 (left): Specific tank volumes in 1990 for 10 Western European tractor brands [4.26]. Power acc. ECE R 24

Specific tank volumes of a European tractor program of 2016 (excluding narrow track tractors) are plotted in **Fig. 4.25**.

Such tank volumes can be evaluated by calculations for heavy tillage, for example with the following parameters:

- Tractor with 150 kW rated power
- 12 hours of very heavy work
- Average use of 80% rated power
- Specific fuel consumption of 212 g/kWh
- Diesel fuel density of 0.84 kg/l

This would result in a fuel consumption of 0.212 kg/kWh·150 kW·0.80·12 h = 305.3 kg = 363.4 l fuel = **2.42 l per installed kW**.

The following values are recommended for standard tractors in [4.27]:

- Light duty .... 1.5 l/kW*)
- Medium duty .... 2.0 l/kW
- Heavy duty .... 2.5 l/kW

*) Rated power ECE R 24 (defined with fan)

Burning 1 liter of diesel fuel produces as much as 2.7 kg CO2 [4.24, 4.28].

Investigations have been carried out on practical fuel consumption and CO2 emissions for field operations [4.28, 4.29].

A fuel economy guideline for engine working points is listed in **Table 4.5** for an evaluation of published test results.

Duty cycles for an average use over a longer period of time can be of interest to evaluate average fuel consumption, average emissions, average noise levels, etc.

The duty cycle of **Table 4.6** was used by Deutz [4.30, 4.31] to rate the benefits of an infinitely variable cooling fan regarding average fuel consumption of Deutz tractor engines over one year (average power use 40.2% of rated power). The forecasted fuel savings based on this duty cycle could be confirmed well by practical fuel consumption measurements in Germany.

A calculation of the average fuel consumption results on this basis in **about 100 g per installed engine kW hour**. Small tractors may be lower, large ones rather higher.

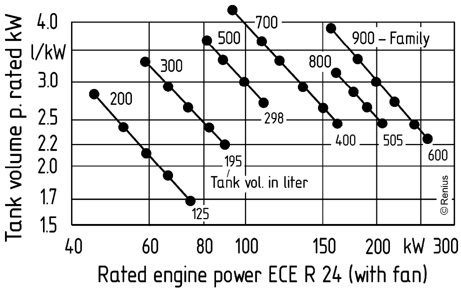


Fig. 4.25: Map of tank volumes of the complete AGCO-Fendt tractor line, as of 2016.
Data courtesy AGCO-Fendt

Table 4.5: Guideline for excellent tractor fuel economy: SFC 1 and SFC 2 for the engine (standalone with fan), SFC 3 and SFC 4 for simplified measurements at the PTO shaft. Source: Author

Power range	SFC 1 g/kWh	SFC 2 g/kWh	SFC 3 g/kWh	SFC 4 g/kWh
37-56 kW	218	240	240	264
56-75 kW	214	235	235	259
75-130 kW	208	229	229	252
130-250 kW	202	222	222	244
>250 kW	196	216	212	233

Definitions:

SFC 1: Lowest specific fuel consumption – engine
SFC 2: Full load specific cons. @ rated speed – engine
SFC 3: Lowest specific fuel consumption – PTO
SFC 4: Full load specific cons. @ rated speed – PTO

Assumptions:

SFC 2 = 1.10 x SFC 1
SFC 3 = 1.10 x SFC 1 (37–250 kW)
SFC 3 = 1.08 x SFC 1 (above 250 kW)
SFC 4 = 1.10 x SFC 2 (37–250 kW)
SFC 4 = 1.08 x SFC 2 (above 250 kW)

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Table 4.6: Deutz five-point duty cycle for average tractor use over one year [4.30, 4.31].
T - Torque, T_{nom} - rated torque, n - rot. speed

T/T_{nom} %	n/n_{nom} %	$T \times n$ %	Time %	Typical tractor operations
88	95	83.6	31	Heavy pull and PTO operations of all kinds
48	85	40.8	18	Moderate field pull, also with PTO, field transports
40	53	21.2	19	All typical operations with reduced engine speed
15	100	15.0	20	On-the road movements at high speeds, low loads
0	40	0	12	All portions of idling

4.7.2 Diesel fuel specifications and altitude corrections

Classic diesel fuel has been used over a long period of time with relatively high sulfur content as one typical fraction of crude oil [4.32]. Even low concentrations of 500 ppm (0.05%) are undesirable:

- *Sulfur burns to SO_2, a harmful emission*
- *Sulfur disrupts exhaust aftertreatments*

Both items are very important and initiated the development of *ultra-low sulfur diesel* (ULSD). It has been legislated for non-road engines to enable advanced emission control systems, limited to 10 ppm for EU and 15 ppm for the USA (Tier 4).

Table 4.7: Basic specifications of diesel fuel from DIN EN 590 (2014). Caloric value per liter is 9.7 kWh, creating 2.65 kg of CO_2

Criteria	min.	max.
Cetane number	51	
Density (15 °C), kg/m^3	820	845
Sulfur content, mg/kg (=ppm) (within EU as of 1.1.2009)		10
Water content, mg/kg		200
Viscosity, mm/s^2 (40°C)	2.0	4.5
Biodiesel volumetric %		7.0
Manganese content, mg/l (within EU from 1.1.2014)		2.0
Filtrability see standardization for "climate zones"		

ULSD requires additional refining procedures and lubricating additives, creating additional costs – one reason that ULSD is not available in many developing countries.

Specifications of diesel fuel are standarddized, example see **Table 4.7**. In addition, filterability must be defined. Most engines can also burn biodiesel. Pure rapeseed oil (DIN 51605) requires modifications, best results are achieved under high loads.

Special conditions. Diesel engines with natural aspiration are sensitive to ambient air temperatures but much more to high altitudes [4.33], **Fig. 4.26**.

The injection rates must be reduced in order to prevent malfunctioning burning, which can damage the engine and can create a dramatic increase in pollutants, mainly of particulates.

The example of Fig. 4.26 for 3000 m height above sea level and 20°C environmental air temperature requires a reduction of the injection rate by 25% to secure proper burning. *The customer must be prepared for a power reduction of also 25%.*

Such a reduction is usually not necessary for tractors with turbocharged engines.

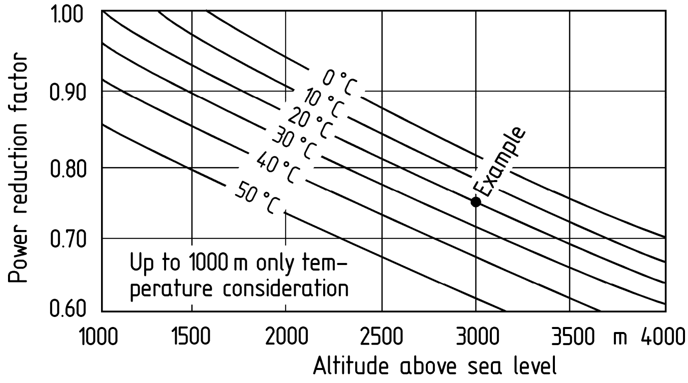

Fig. 4.26: Correction factor for diesel engines with natural aspiration: Adjustment of fuel injection to ambient temperature and local altitude Courtesy Deutz

4.8 Diesel engine installation and lubrication

Motorization concept. A complete tractor line is usually formed by tractor families, as outlined in chapter 1.6.

Common outer engine dimensions for all models within the family save costs.

Engines with three, four, and six cylinders are used for tech levels III, IV, and V.

The best gradation of the nominal engine power of neighbouring models should be based on a *geometrical sequence*. This is best represented on a logarithmic scale: Ratios become distances, Fig. 4.25.

The 4-cylinder engine was and is the most popular one in many countries, growing over the decades in rated power by increased displacements and mean effective pressures. Specific power levels are between about 15 and 40 kW per liter while typical displacements range from about 3 to 6 liters (Table 4.2).

Power characteristics. They can be modified in a wide range – mainly in the case of electronic engine control.

Regarding the nominal power, tractor diesel engines usually do not work at their potential maximum speeds although the specific costs per kW would be lower if they did. The main reasons are:

- Better fuel economy
- Lower noise level
- Lower transmission and PTO ratios
- Higher transmission efficiency

Another market demand is a solid torque rise (also called torque back-up), **Fig. 4.27**.

Two reasons can be named:

- “Stiff” reaction when loads increase
- Speed range of constant power

The higher the value, the lower the possible nominal power. Some companies offer a boost to use the shaded area of Fig. 4.27 under specific operating conditions.

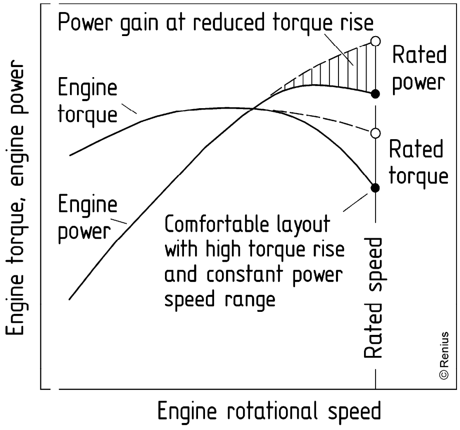

Fig. 4.27: Higher engine torque rise results in lower rated power

Diesel engine installation needs a lot of development steps, mainly for block chassis concepts, **Table 4.8**. Temperature control under full power and hot ambient air is the most important issue [4.33, 4.34].

Table 4.8: Activities of diesel engine installation

- Determine rated engine power and rated speed
- Determine torque rise and related speed drop
- Codify power definition (with or without fan)
- Decide “block chassis” or “engine in silent blocks”
- Harmonize connection details with transmission
- Block chassis: Ensure engine block durability
- Decide air cleaner concept (press. drop < 3 kPa), observe air speed limits at the bonnet grid
- Determine required engine cooling capacity
- Determine other cooling capacities, e. g. EGR
- Design external cooling package and fan drive
- Design exhaust system and after-treatment
- Test exhaust system regarding vibrations
- Design position and drive of all auxiliaries (consider possibility of driving by PTO life shaft)
- Design fuel (and urea) tank system
- Install electronics for engine diagnostics
- Connect engine electronics with tractor CAN bus
- Optimize air flow and temperature distribution
- Observe fuel temperature limits
- Measure and optimize engine noise
- Secure meeting of emission limits
- Test complete system under field conditions

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Engine cooling must be designed *to get rid of about 30% of the fuel energy for the maximum engine power at ambient temperatures of up to about 40°C.*

Precise specifications depend on engine size, degree of encapsulation, portion of engine oil cooling, and local temperatures.

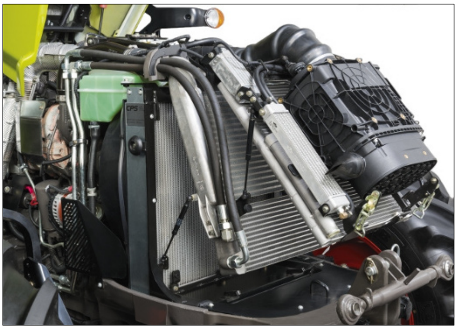

Fig. 4.28: Cooler-filter package with swing-out for maintenance. Courtesy CLAAS

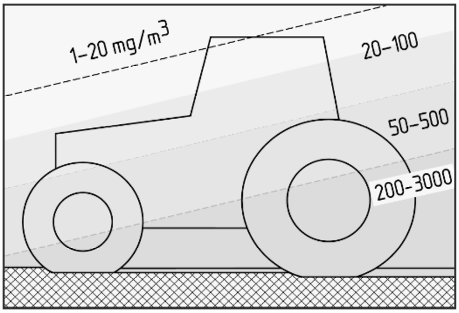

Fig. 4.29: Dust concentration zones for dry field tillage, guideline. Source: *Kutzbach* [4.35]

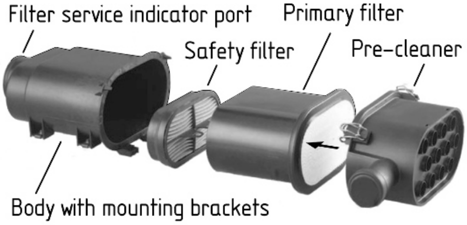

Fig. 4.30: Modern engine air filter system with paper cartridge. Courtesy Donaldson Company

The main cooling can be done directly by air (famous Deutz engines) or indirectly by air via water-based liquids.

Air cooling is still popular in some very specific cases but has disadvantages:

- Limited power density
- High specific costs (light metals)
- Difficult to meet exhaust emission limits
- Poor ability to heat the cab
- Large engine width
- Uncomfortable tone of noise

Heat transfer in **water coolers** is supported by a fan, which can create high parasitic energy losses during high speeds.

The following requirements are typically for the upper technology levels:

- High fan efficiency (aerodynamic design)
- Automatic fan speed control (at part load)
- Reversible fan speed (cleaning mode)
- Easy access to cooler package (handling)
- No recirculation of heated air (efficiency)

The main cooler is usually combined with several others, **Fig. 4.28** with a well designed gradation of the temperature levels.

Engine air cleaning is necessary for limiting wear mainly for the sliding combination of piston and cylinder. The dust concentration around a tractor during dry off-road operation is pretty high, **Fig. 4.29**.

Oil bath cleaners were used for many decades [4.36], in Europe until about 1970. They are very robust and have low maintenance costs; however, they do not work as perfectly as paper filter cartridges which achieve separation rates of 99.5 to 99.9 %.

Engine air throughput increases by about a power of three on the scale but all areas and cross-sections only by a power of two, which results in over proportional increased openings along with increased power.

Hence, it is a challenge to develop air flow dimensions and filter cartridges with tolerable dimensions. The main filter is usually combined with a precleaner, **Fig. 4.30**, for example by a cyclone separator.

Lubrication and engine oil circuit. The central function of engine lubrication is to produce an oil film separating sliding elements and reducing wear on rigid contacts (mixed friction). This would, however, need only very small oil flows.

The second important duty is to carry heat away (from friction and burning). This requires a much higher flow than for the wear reduction. Pumping energy can be saved by temperature-adjusted pump displacement, as already used in some cases.

Diesel engine lubrication fluids perform the following duties:

- Lubrication by hydrodynamics (viscosity)
- Minimizing contact wear (additives)
- Cooling (thermodynamics)
- Preventing internal corrosion (additives)
- Internal cleaning (additives)

Modern diesel engines of the upper technology levels require **very special oil specifications** due to:

- Fuel with extremely low sulfur content
- Extremely high injection pressures
- Particulate filters
- Chemical exhaust aftertreatment
- Expanded oil change intervals

A new generation of diesel engine oils has been developed, meeting, for example, the quality specification API-CJ-4 of the "American Petroleum Institute" [4.37].

Micro oil filters are a general must, being replaced together with the oil (250–600 h) or on demand (electronic control).

Oil viscosity is the basic parameter for lubrication representing the internal resistance against shear motion. It is extremely influenced by temperature, **Fig. 4.31**.

The higher the viscosity, the better the hydrodynamic load capacity but also the higher the energy losses due to shear forces.

Viscosity is defined in two forms:

1. Kinematic viscosity ν [mm²/s, m²/s]
2. Dynamic (absolute) viscosity η [Ns/m²]

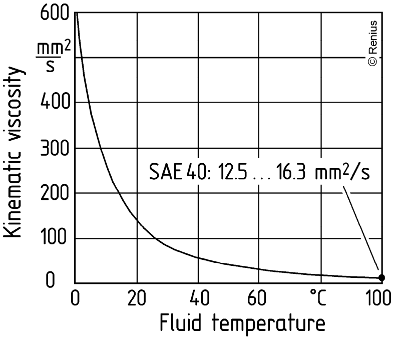

Fig. 4.31: Viscosity of a pure mineral oil (after *Eckhardt*, cited in [3.31])

The standard SAE J 300 is in use for kinematic viscosities of IC engines [4.38]. Modelling of journal bearings and other sliding surfaces needs the dynamic (absolute) fluid viscosity, first derived by Isaac Newton [4.39]. "Newton fluids" develop a linear speed between moving plates, **Fig. 4.32,** which is widely the case for engine lubrication.

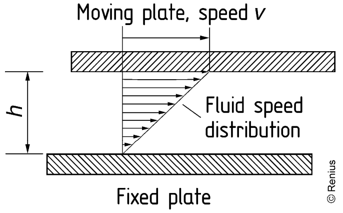

Fig. 4.32: Shear speeds of a "Newton" fluid

The dynamic (absolute) viscosity η is the "fluid factor" of the fluid shear stress τ:

$$\tau = \eta \cdot \frac{v}{h} \tag{4.9}$$

where v and h are defined in Fig. 4.32.

The kinematic viscosity ν is

$$\nu = \frac{\eta}{\rho} \tag{4.10}$$

where ρ is the mass density of the fluid.

Viscosity classes for tractor engines are: SAE 15W-40 / SAE 10W-40 / SAE 5W-40 covering the high temperature viscosity of SAE 40 and the low temperature viscosity of SAE 5W / 10W / 15W – by additives.

4.9 Energy balances and alternative powertrain concepts

Typical total off-road efficiencies from diesel fuel to implement draft power are about 20–25% – thus moderate. If we assume an average burning energy of 11.8 kWh/kg diesel fuel, engine efficiency is

$$\eta = 84.7 / b_{\mathrm{e}} \tag{4.11}$$

where b_{e} = engine consumption in g/kWh.

Example: $b_{\mathrm{e}} = 200$ g/kWh, $\eta = 0.42 = 42\%$.

40% engine efficiency is, for example, equivalent to a specific consumption of 212 g/kWh, which is offered by modern diesel engines of the upper technology and power levels for engine power loads over about 50%, as, for example, realized in Fig. 4.22 and 4.23.

A complete power balance is developed on this basis in **Fig. 4.33**. The highest losses can be identified for the ground drive and for the hydraulic PTO (value 80% for LS systems with power beyond).

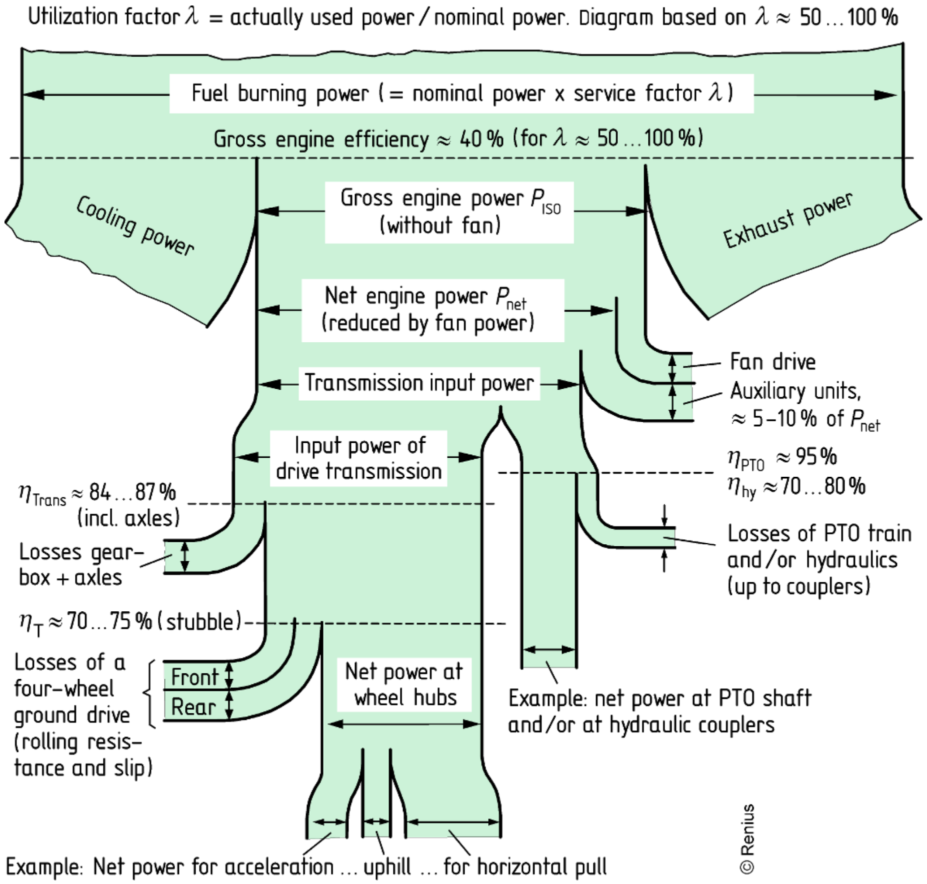
Utilization factor λ = actually used power / nominal power. Diagram based on λ ≈ 50 ... 100 %
Fuel burning power (= nominal power x service factor λ)
Gross engine efficiency ≈ 40 % (for λ ≈ 50 ... 100 %)
Cooling power
Exhaust power
Gross engine power P_{ISO} (without fan)
Net engine power P_{net} (reduced by fan power)
Fan drive
Auxiliary units, ≈ 5–10 % of P_{net}
Transmission input power
Input power of drive transmission
η_{PTO} ≈ 95 %
η_{hy} ≈ 70 ... 80 %
η_{Trans} ≈ 84 ... 87 % (incl. axles)
Losses gear-box + axles
Losses of PTO train and / or hydraulics (up to couplers)
η_{T} ≈ 70 ... 75 % (stubble)
Losses of a four-wheel ground drive (rolling resistance and slip)
Front
Rear
Net power at wheel hubs
Example: net power at PTO shaft and / or at hydraulic couplers
Example: Net power for acceleration ... uphill ... for horizontal pull
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Fig. 4.33: Power balance of a tractor with stepped transmission and utilization factor λ above 0.5, driving within a range of nominal speeds of about 5 to 15 km/h and delivering some PTO power. Practical λ values for a manual transmission and heavy loads are about 0,70-0,85. Transmissions with automatic "full power control" can achieve values of about 90-95%.

Alternative powertrain concepts have been studied since the beginnings.

It was not clear around 1900 whether the internal combustion engine (IC engine) would prevail over the battery-powered electric motor, which was often more reliable. Likewise, steam-powered tractors achieved great popularity from about 1890 to 1920, often featuring better reliability than IC engines (mainly in the US).

Updated steam power engines could today offer much better efficiencies than in the past owing to higher pressures, higher temperatures, and more efficient condensers [4.40]. They could use almost every fuel and produce very low emissions owing to the external burning. Steam power would allow a vehicle drive-off from standstill without clutches or CVTs, simplifying the transmissions as compared with those of present tractors. Steam power creates only very low noise levels.

The main disadvantages are, however:

- High degree of technology (costs per kW, closed fluid-steam-fluid circuits)
- Large required space (condenser)
- Moderate full power efficiencies
- Moderate dynamics

Several prototypes have been presented; an interesting modern development from IAV (Germany) has been published in [4.41]; however, without a breakthrough.

Gas turbines were studied early as prototypes for road vehicles [4.42].

Their main advantages are the extremely high power density, continuous burning, smooth torque output, excellent durability, and low noise level.

Efficiency is below that of diesel engines; they need “more transmission” and are much higher in initial costs per kW.

Direct diesel-electric drives also offer some interesting advantages – mainly CVT quality. Electric energy transfer to implements is considerably superior in efficiency to a hydrostatic power transfer with some benefits for auxiliary units as well [4.43].

The main disadvantages are, in comparison with hydromechanical power split CVTs, higher initial costs, higher weights, and slightly lower efficiencies.

The technologies of electric drives, however, show a positive trend regarding the automotive industry. High voltage applications (e. g. 700 V) allow new energy-efficient solutions. Safety issues have been addressed by an ISO standard [4.44].

Diesel-electric drives with power split could achieve at least the efficiencies of the best present hydro-mechanical solutions.

They promise improved auxiliary drives, low noise levels, boost functions and high durability. In 2019, John Deere announced a first development, but a quick replacement of all present CVT designs is not expected.

Fuel cells offer the highest efficiency to turn chemical energy into electricity, mainly when using nitrogen, with the advantage of zero emissions [4.45].

Present problems are

- Technology not yet reliable for tractors
- Still expensive
- Limited storing capacity for hydrogen
- Availability and costs of nitrogen

In 2009, CNH presented a study, **Fig. 4.34**.

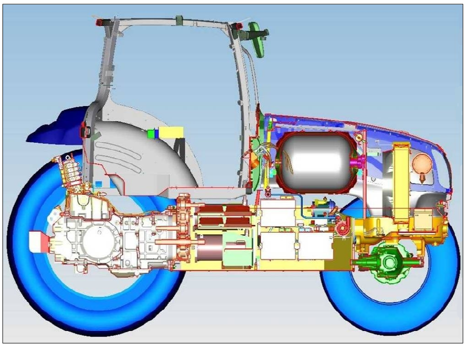

Fig. 4.34: Agricultural tractor with hydrogen fuel cell, CNH study, SIMA 2009. Courtesy CNH

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Other suggested publications

Chapter “tractor engines and transmissions” within the annual German Yearbook Agricultural Engineering, see notes on page 179.

5 Tractor transmissions

5.1 Introduction, history, and global market demands

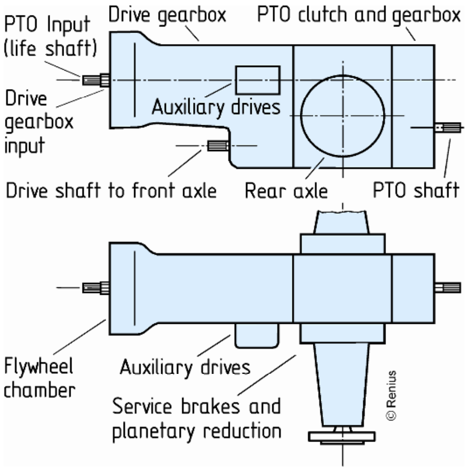


Fig. 5.1: Typical configuration of a tractor transmission (“transaxle”), typical example for upper technology levels

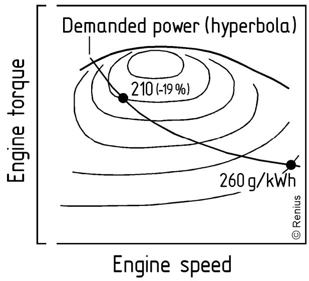
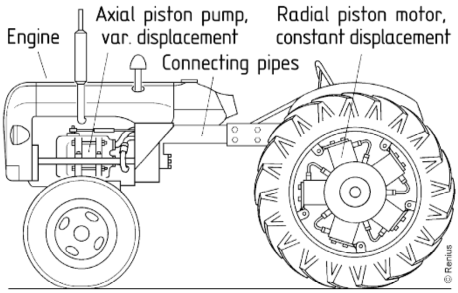


Fig. 5.2: Engine working points: Strategy of optimized fuel economy. Benefit of example = 19%



Importance and general functions. The transmission with included rear axle, also called “transaxle”, **Fig. 5.1**, is the most important tractor component, representing about 25–30% of total tractor cost.

It contains not only the gearbox but also the complete rear axle with differential, differential lock, rear brakes, power take-off system (PTO), auxiliary drives, and the output gearing and shaft for the front drive.

The main function *is to allow the use of as many engine working points as possible – in the ideal case all of them infinitely.*

The sub-function *is to realize power take-off (PTO) interfaces for implements and drives of auxiliary components.*

The **ideal gearbox** offers a continuously variable ratio control; see the famous early proposal of an internal power split CVT of *L. Renault* (1907) for passenger cars [5.1].

An efficient CVT allows the use of every engine working point with the absolutely best possible fuel economy, **Fig. 5.2**.

Several infinitely variable transmissions have been built in study units or prototypes after World War II.

The Silsoe Research Tractor with hydrostatic transmission gained international attention in 1954, **Fig. 5.3** and Table 1.1 [5.2].

In 1958, J.I.Case introduced a tractor with a hydrodynamic torque converter (with a locking clutch) [5.3]. ZF-P.I.V., J. Deere, and other companies developed prototypes with variable steel chain drives [5.4].

In 1988, the Munich Research Tractor was presented with a P.I.V. steel chain drive, two additional ranges forward and one reverse range (vehicle see chapter 6).

Fig. 5.3 (left)**:** NIAE Research Tractor with infinitely variable hydrostatic ground drive, presented in 1954 by National Institute of Agricultural Engineering, Silsoe, UK [5.2]

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But no concept could achieve series production. The initial costs would have been much higher and the efficiency much lower as compared with the level of stepped transmissions within that period.

Years later, in 1966, the German company Eicher, along with the British company Dowty, introduced the first hydrostatic gearbox for a standard tractor [5.4]. 71 units were produced 1966–72.

In 1967, International Harvester (USA) presented tractors with hydrostatic gearboxes from a co-operation with Sundstrand [5.4, 5.5]. IH sold about 15,000 units (estimated for all models including industrial versions).

The development now concentrated more on stepped transmissions with different priorities in Western Europe and the USA.

IH presented the first **Hi-Lo power shift** in 1954 [5.6, 5.7], **Fig. 5.4**. The idea was to improve both comfort and productivity by doubling the conventional speeds with a power shifted Hi-Lo unit splitting the 5/1 speeds into 10/2 by fingertip. When the clutch is closed, the unit rotates as a block 1:1. When the clutch is released, the planetary carrier speed slows down to be automatically connected with the housing by the free-wheeling. The unit now works like a countershaft reduction with a slightly lower output speed, to meet, for example, a higher implement draft requirement (lower speed allows higher axle torque). Similar designs from other tractor manufacturers followed (see also [5.7]):

- 1955 Minneapolis Moline “Ampli Torque”
- 1957 Allis-Chalmers “Power Director”
- 1962 Massey Ferguson “Multi Power”
- 1962 Oliver “Hydra Power”
- 1963 IH “Torque Amplifier II”
- 1967 Deere “L-S transmission”
- 1967 ZT 300 “ULS” (GDR)
- 1969 Ford “Dual Power” and others.

Highest production numbers are known from IH, John Deere, Ford, and MF.

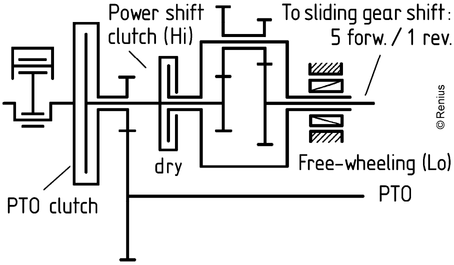


Fig. 5.4: IH “Torque Amplifier” of 1954, the first Hi-Lo power shift for tractors [5.6, 5.7]

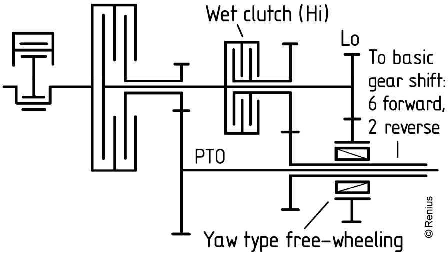


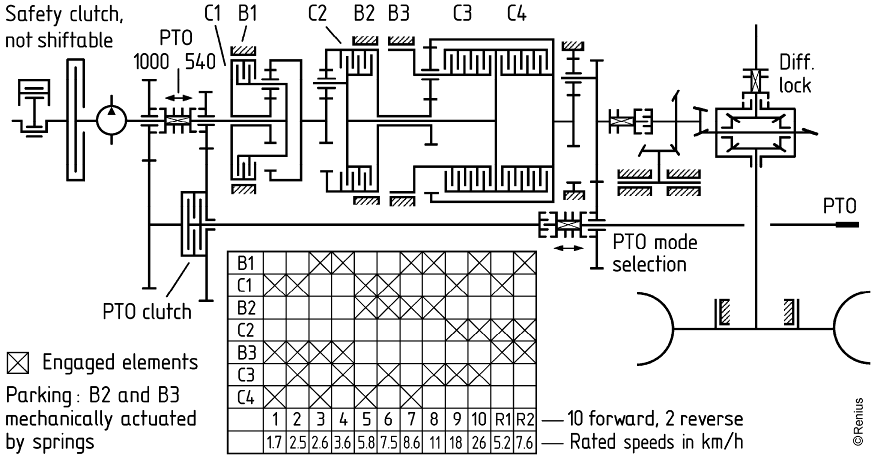
Fig. 5.5: Massey Ferguson “Multi Power”, presented in 1961, in series in 1962 (MF 65) [5.8]

MF used a countershaft design with an axial overrunning clutch, **Fig. 5.5** [5.7, 5.8].

Early versions with **more than two power shifted speeds** (in brackets) were:

- 1958 Ford “Select-O-Speed” (10/2) [5.9, 5.10]
- 1963 J. Deere “Power Shift” (8/4) [5.11]
- 1968 Oliver “Over-Under” (3) [5.12]
- 1970 Case “Agri King” (3/1) [5.13]
- 1971 David Brown “Hydra Shift”(4) [5.14]
- 1975 Allis-Chalmers (3 or 6) [5.15]
- 1976 Massey Ferguson (3) [5.16]
- 1978 Deutz DX230 - ZF T6600 (3/1) [5.17]

About 500 units of the ZF T 6600 were produced; however, a breakthrough of partial power shifts started in Western Europe only in 1989 with the 4-speed power shift of the Case IH “Maxxum” (4/4), followed in 1991 by Ford (4) and Renault (3), in 1992 by John Deere (4) and MF-Gima (4), in 1993 by Fendt-ZF (4), in 1995 by Deutz-ZF (3, 4), in 2005 by MF-Gima (6), and in 2006 by CLAAS-Gima (6).



B1			☒	☒			☒	☒		☒		☒
C1	☒	☒			☒	☒			☒		☒	
B2					☒	☒	☒	☒				
C2									☒	☒	☒	☒
B3	☒	☒	☒	☒							☒	☒
C3		☒		☒		☒		☒	☒	☒		
C4	☒		☒		☒		☒					
	1	2	3	4	5	6	7	8	9	10	R1	R2
	1.7	2.5	2.6	3.6	5.8	7.5	8.6	11	18	26	5.2	7.6

Fig. 5.6: First "full power shift" tractor transmission Ford "Select-O-Speed", USA 1958 [5.9, 5.10]. Speeds 2 and 3 almost identical. "Power Shift" of John Deere (1963–82) see chapter 1.3

The Ford "Select-O-Speed" worked with three planetary sets for 10/2 speeds forward/reverse, **Fig. 5.6** [5.9]. It had a direct simple shift valve while John Deere used a pilot valve system in its 8/4 "Power Shift" (1963, chapter 1.3). The variety of transmission functions and power levels has increased over the last decades, hence the author proposed in 1999 a classification of global transmission specifications by technology levels, **Table 5.1** ([5.20] updated).

The success of John Deere's Power Shift led to its enhancement in 1982 to a 15/4 version [5.17, 5.18]. Other companies followed, for example, Case IH with a 18/2 full power shift (1987 for Magnum) [5.19].

The challenge of developing new transmission systems globally is to cover the very large variety of demanded functions by a limited number of components and parts. This requires outstanding professional long-term product planning.

Table 5.1: Worldwide demanded tractor transmission specifications by five technology levels [5.20]

		Speeds, km/h		No. of speeds		Type of shift	PTO rpm	Final drive	Service brakes	Lubrication	IT-level	Typical markets
		forward	rev.	forw.	rev.							
Technology level	I	2–20 (25)	3–8	5–8	1–2	Slid, Coll	540	Bull gear or planet.	dry & wet	by splash	no electronics	Small tractors in Africa, India, China, South Am.
	II	2.2–30 (32)	3–15	8–12	4	Coll, Syn	540 (1000)		Disc brakes, wet			
	III	(0.5) 2.4 −30 (40)	3–15	12–24	4–12	Syn, Hi-Lo Syn shuttle	540, 1000, (1300) + opt. eco-speeds	Planetary		Oil pump and filter	low	South and East Europe
	IV	(0.3) 2.5 −50 (60)	2–20	16–32 (64)	16	Syn, Part. PS Power shuttle					high	Large tractors in premium markets
	V	2.5–50 (60)	3–20	18–24	6–12	Full PS, auto					very high	
		0–50 (60)	0–20	Infinitely		CVT, auto						

Slid = sliding gear, Coll = collar shift, Syn = synchroshift, Hi-Lo = Hi-Lo power shift, Syn shuttle = synchron. reverser, Part. PS = part. power shift, Power shuttle = power shifted reverser, Full PS = full power shift, CVT = continuously variable transmission, auto = automatic shift.

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Tractors of technology levels III/IV/V (Table 5.1) can be considered to be the most important section in sales.
At the same time, their specifications are more detailed than those for I and II.

For these reasons, general market demands for tractor transmissions of technology levels III/IV/V are specified in **Table 5.2**.

The speed concept of a tractor transmission has to meet practical requirements of the local market. They can differ from area to area. A range from 2.5 to 30 km/h is, for example, not sufficient for Western Europe for technology levels III/IV/V, but may be accepted in some other markets. **Table 5.3** gives a guideline for typical agricultural speeds from a Western European view.

A general rule is that the speed level used increases with the tractor power.

The lowest speed is, for example, required for very slow urban operations with PTO and for planting by hand (e. g. 200 m/h with reduced engine speed).

Table 5.3: Typical working speeds of standard tractors defined for rated engine speed

	km/h
Heavy sub soiling	(3) 4–6
Power tilling (PTO)	3–6
Plowing	(4) 5–10
Heavy cultivating	5–8
Shallow cultivating	7–12
Spreading of manure (PTO)	4–8
... of mineral fertilizer (PTO)	10–15
Urban operations	from 0.5
Planting by hand (vegetables)	0.4–1
Potato planting (PTO)	3–8
Sowing (PTO)	6–15
Spraying (PTO)	10–15 (20)
Mechanical weeding	(3) 4–8
Mowing (PTO, rotary type)	8–15
Pick-up pressing (PTO)	4–10
Front end loading forward	4–8
Front end loading reverse	5–10
Transports	10–30, 40 (50, 60)

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Table 5.2: Market demands for stepped tractor transmissions of technology levels III and IV

1 Forward speeds

- Nominal speeds according to market level; min. (2) 2.5–30 km/h, maximum (0.4) 2–40 (50, 60) km/h
- Ratio of neighboring speeds 1.16...1.22 for main working range (4...12 km/h). Higher ratios below 4 km/h and above 12 km/h tolerated

2 Reverse speeds

- Bandwidth nom. speeds min. (2) 3...15 (20) km/h
- Ratio of neighboring speeds 1.2 ...1.3, low values in the case of important field reverse operations
- Reverse speeds corresponding to forw. speeds

3 Speed shift

- Shiftability: Safe, fast, comfortable, precise
- Low number of levers, low shift forces, adequate lever positions and shift travels, shift pattern easy to understand. Partial power shift
- Onroad: "One-lever-shift" possible for starting with a gear below 10 km/h rated speed
- Comfortable shifting forward-reverse ("shuttle")

4 Pedals: Adequate pedal forces (clutch, brakes)

5 Rear power take-off (chapter 5.4)

- PTO type 1 (ISO 500): 540 rpm at about 90% rated eng. speed (relation not standardized)
- PTO type 2 or 3 (ISO 500): 1000 rpm at 95–100 % rated engine speed (relation not std.), shiftable from one shaft type or with easy shaft change;
- Many markets require additional eco speeds: e. g. 750 and 1250 rpm to run 540 and 1000 rpm at lower engine speed
- PTO type 3 (ISO 500): 1000 rpm for high power
- PTO type 4 (ISO 500): 1300 rpm for very high power with larger shaft diameter
- PTO rpm proportional to travel speed (as seldom option): often 3–9 rotations per 1 m travel distance

6 Auxiliary drives should be possible, usually offered by access to the PTO life shaft

7 Rear axle

- Adequate clearance and track (chapter 3)
- Differential lock, easy to disconnect
- Brakes meeting traffic regulations (chapter 3)

8 Durability, repairs, maintenance, efficiency

- Transmission B10 life: 8,000...10,000...12,000 h, (increases with tractor power)
- Oil circuit by transmission oil pump with filter
- Spare parts: Low demand (lifetime brakes)
- Maintenance demands as low as possible: Oil change intervals above 2000 h, filter replacement on request (pressure drop gauge)
- Total transmission efficiency (input shaft to wheels) at least 85 % at 50–100 % load for nominal speeds of 5–15 km/h, not much decreasing at higher speeds.

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5.2 Speed configuration for stepped transmissions

Practical speeds (Table 5.3) must be weighted by time portions in order to decide priorities of the planned transmission speeds. Unfortunately, the demanded speeds vary not only from market to market but also from farm to farm.

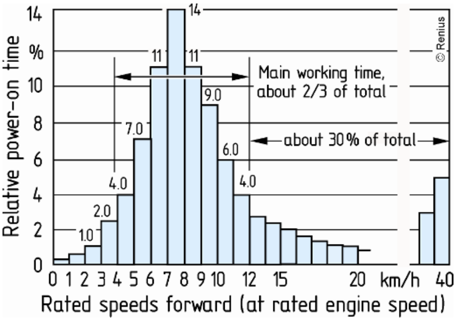


Fig. 5.7: Average distribution of tractor speeds. Updated estimation of [5.21] for Western Europe, typical for tractors of about 80–150 kW

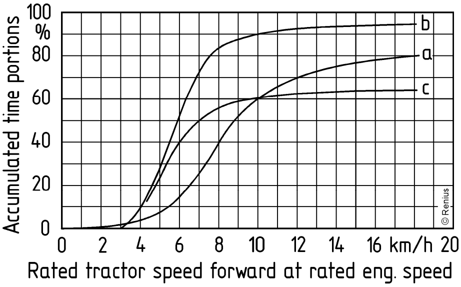


Fig. 5.8: Estimated average accumulated time portions of tractor speeds for three typical cases:
a - Western Europe, values from Fig. 5.7
b - Mainly very heavy tillage, no transports
c - Low power, heavy tillage, transports important

Fundamentals are therefore only possible as a very rough guideline. **Fig. 5.7** shows an estimate of the average use of tractor speeds in Western Europe. This is an update of [5.21] as the speeds have meanwhile slightly increased.

The larger the tractor, the lower the specific mass (kg/kW) and the higher the speeds used for efficient traction. For this reason the distribution may be shifted again for standard wheel tractors 0.5–1.0 km/h to the right for about 200–250 kW and 1.0 km/h for above 300 kW rated power.

Other markets can deviate, **Fig. 5.8**.

Case a is identical with Fig. 5.7.

Case b is typical for tractors which are mainly used for heavy tillage of dry soils.

Case c is typical for many developing countries such as India and those in Africa.

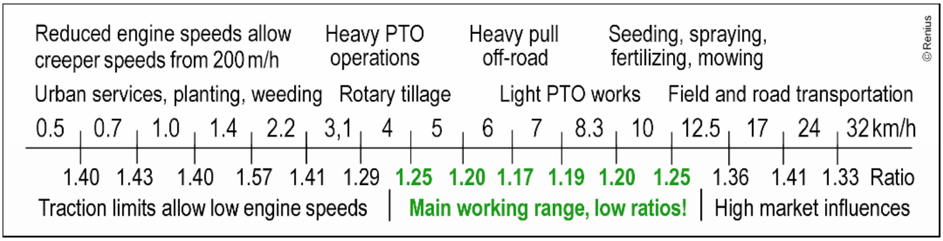
A first speed concept uses a geometric series with the ratio C of neighboring speeds:

$$C = \sqrt[x-1]{\frac{V_{\max}}{V_{\min}}} \tag{5.1}$$

where x is the given number of speeds, $v_{\min}$ the lowest given speed and $v_{\max}$ the given top speed.

Example: Speed bandwidth 0.5–32 km/h, 16 speeds, calculated step ratio = **1.32**. This value is too high within the main working range of a tech level III transmission. It demands a speed concentration within 4–12 km/h [5.22], **Table 5.4**, or more speeds.

Table 5.4: Example of a target for a transmission with 16 speeds for technology level III. Given range: nominal speeds 0.5–32 km/h. Concentration of speeds within the 4–12 km/h range


Reduced engine speeds allow creeper speeds from 200 m/h
Heavy PTO operations
Heavy pull off-road
Seeding, spraying, fertilizing, mowing
© Renius
Urban services, planting, weeding
Rotary tillage
Light PTO works
Field and road transportation
0.5 0.7 1.0 1.4 2.2 3,1 4 5 6 7 8.3 10 12.5 17 24 32 km/h
1.40 1.43 1.40 1.57 1.41 1.29 1.25 1.20 1.17 1.19 1.20 1.25 1.36 1.41 1.33 Ratio
Traction limits allow low engine speeds
Main working range, low ratios!
High market influences

Rule of thumb: 3-4-5-6-7-8.3-10 km/h.

A speed concentration within the main working range can save gear wheels, causing additional savings of shaft length, oil quantities, case weights, etc. – in the end, money.

The disadvantage of this philosophy is that a doubling (splitting) of the whole speed pattern does not automatically result in even more well-balanced speeds (as can be in the case of a geometric gradation).

Representation of **tractor performance** for a given transmission layout should use a **double logarithmic scale**, **Fig. 5.9**.

The benefits are:

- Points of nominal power on a straight line
- All performance curves have the same size
- Step ratios are represented by distances

The tractor speed v is:

$$v\,[\text{km/h}] = 0.377 \cdot r\,[\text{m}] \cdot \frac{n_{\text{eng}}\,[\text{rpm}]}{i_{\text{total}}} \qquad (5.2)$$

where

r = effective tire radius [m]
n_{eng} = rot. engine speed [rpm]
i_{total} = overall reduction ratio
= engine speed / rear tire speed

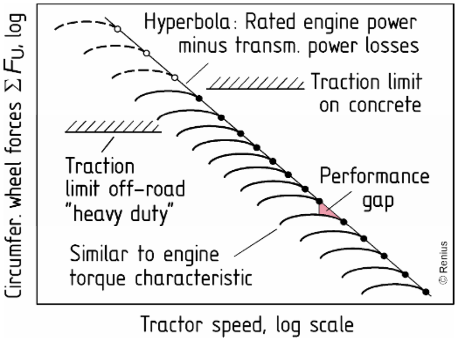
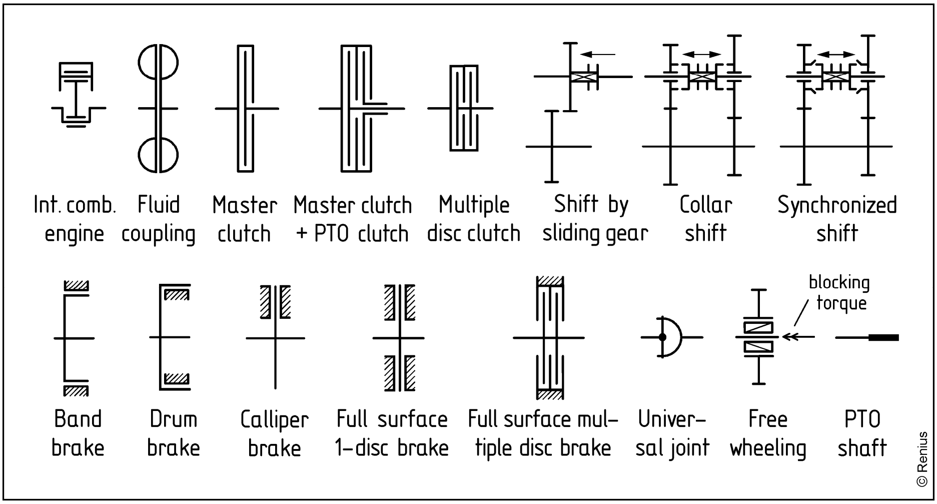


Fig. 5.9: Tractor performance for transmission speeds, excellent overview by log-log scaling

Symbols for transmission elements can help to develop, understand, discuss, and document transmission functions including homologation documents.

The symbols of **Table 5.5** were originally proposed in 1968 in [5.7] since then having been used for all transmissions presented by the author, and used in university lectures, seminars, and homologation documents as well. They have been well proven over more than 50 years and are applied to all transmission maps of this book.

Table 5.5: Symbols for transmission structures, *Renius* 1968 [5.7], proved in more than 50 years



5.3 Stepped transmissions: how to save gear wheels

The gearbox of stepped tractor transmissions usually has a high number of speeds; thus, a high number of gear wheels influencing costs in and of itself, as well as through the dimensions of surrounding elements. Guidelines are shown in **Fig. 5.10**. Sixteen gear wheels for eight speeds forward, **version A**, are not economic.

The **principle of forming groups** reduces gear wheels and costs, **version B**.

Four basic speeds are combined with a two-speed range gearing L-H creating a total of 4 x 2 = 8 speeds. Such versions with six pairs of gear wheels have been produced in large numbers and are still of interest for technology levels I and II. A single overlapping of the speeds of the ranges L and H results in a speed concentration within the main working range of 4–12 km/h (Fig. 5.12 opposite page).

Looking for further reductions of gear wheels for even lower costs, it is possible to generate eight speeds even with only five pairs of gear wheels, **version C**. Transmissions of this kind were also produced in high volumes, mainly by Deutz and Ford, and are still of high interest for the technology levels I and II. A certain disadvantage is that they have some restrictions in choosing the speeds independently. However, CARRARO has adopted this principle for an Indian transmission (Fig. 5.28).

Reducing the number of gear wheels to an absolute minimum [5.23] leads, for example, to **Version D**: Only four pairs of gear wheels, but with major disadvantages:

- A high number of shifting elements
- Complicated shifting sequences
- Four unfavourable single shifting elements
- Only three speeds can be choosen independently, five result automatically

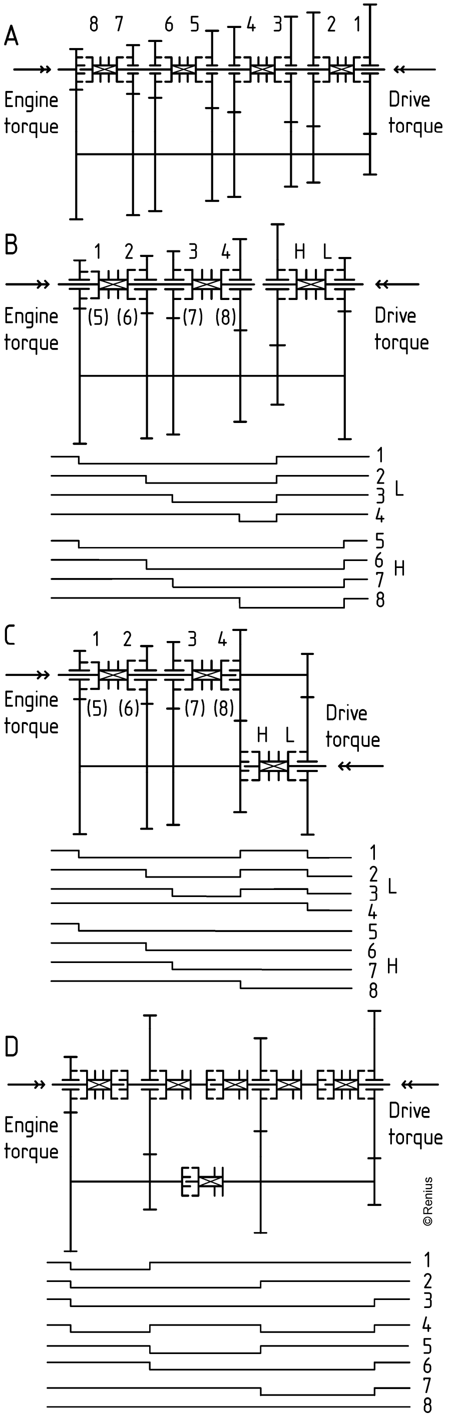


Fig. 5.10: Principles of reducing gear wheels

Stepped transmissions of technology levels III and IV have more than eight forward speeds. As the number of gear wheels influences the basic transmission costs, counting their number is of basic interest.

Fig. 5.11 gives a guideline for these cases with a band for the recommended number of gear wheels versus the speeds for economic structures [5.23].

The border line in the lower part represents the mathematic limit for two-shaft transmissions [5.24], which can be expressed by the equation

$$x = 2^{\left(\frac{z}{2}-1\right)} \qquad (5.3)$$

where

x = number of forward speeds
z = number of necessary gear wheels

Equation (5.3) is only valid for 2, 4, 8, 16, 32, 64, etc. forward speeds. For speed numbers between these figures, the higher number of gear wheels is always valid.
Example: Ten forward speeds need the same minimum number as 16 forward speeds.

An additional evaluation can be carried out for the reverse speeds, **Table 5.6**. A reverse is even possible with only one gear wheel (see, for example, Fig. 5.31).

Tractor transmissions have at least two reverse speeds, mostly more. A basic design rule recommends integrating the reverse gearing with the range gearing as, for example, realized in the transmission of **Fig. 5.12** with four reverse speeds (instead of two). The reverse speeds are slightly faster than the range L speeds, favourable for front end loading.

The nominal speeds of the Deutz transmission TW 50 are plotted in a logarithmic scale. This has the following advantages:

1. Distances represent speed ratios
2. Best overview of basic gears and ranges
3. Best possible readability

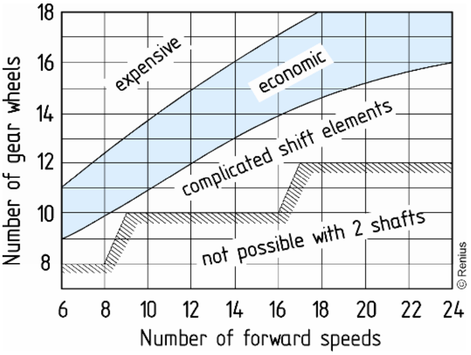


Fig. 5.11: Transmission value analysis by relating the number of gear wheels to the number of forward speeds [5.23]

Table 5.6: Transmission value analysis by counting the number of gear wheels for the reverse speeds [5.23]

Gear wheels for R:	1	2	3	4
Evaluation:	excellent	good	aver.	moderate

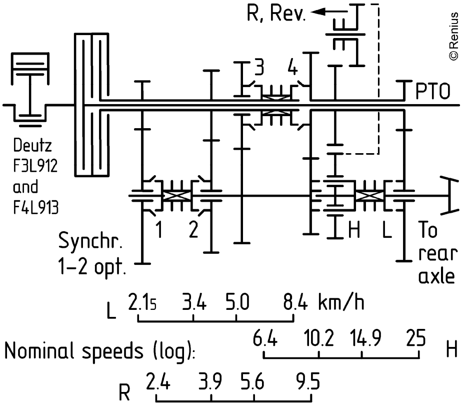


Fig. 5.12: Deutz transmission TW 50, produced 1966-81 in large numbers in Germany [5.14]. Speeds for Deutz tractor D 5206 (1974–81)

Designed by *H. Keienburg,* the Deutz TW50 is a real “Beauty in Engineering”. However, the tolerances of the initial pin shift through the reverse gear were challenging and could be replaced by a design as shown in Fig. 5.96.

The durability of the pinion of the (right) L-range gearing and the pinion of the first gear, optimized through tests with load spectra, was also a piece of art.

5.4 Power take-off (PTO)

5.4.1 Basic PTO configurations

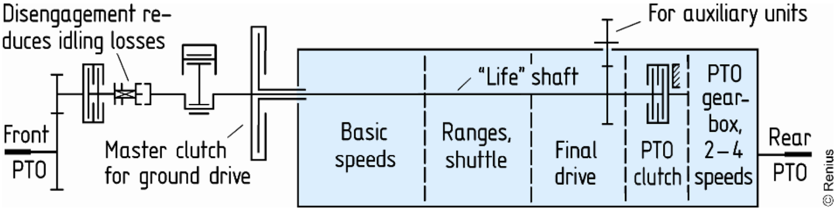


Fig. 5.13: Rear and front power take-off (PTO), a typical configuration for technology levels III–V

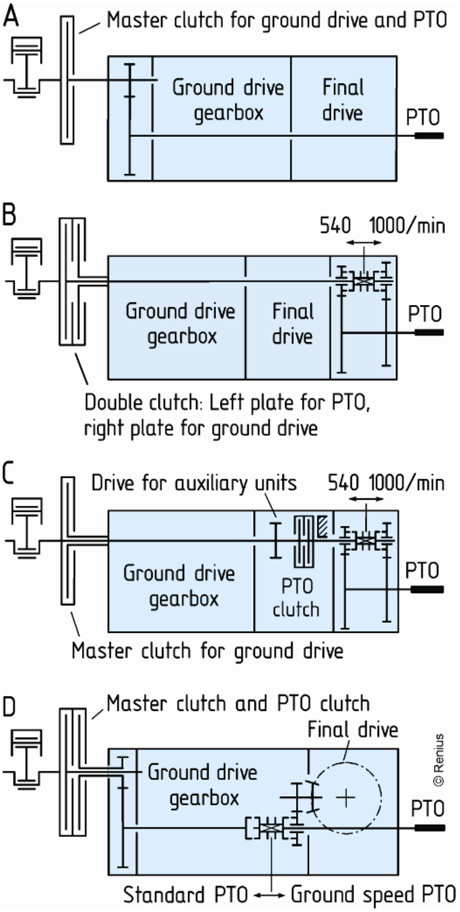

A
Master clutch for ground drive and PTO
Ground drive gearbox
Final drive
PTO
B
540 1000/min
Ground drive gearbox
Final drive
PTO
Double clutch: Left plate for PTO, right plate for ground drive
C
Drive for auxiliary units
540 1000/min
Ground drive gearbox
PTO clutch
PTO
Master clutch for ground drive
D
Master clutch and PTO clutch
Final drive
Ground drive gearbox
PTO
Standard PTO ⟷ Ground speed PTO
© Renius

Fig. 5.14: Basic structures of mechanical rear power take-off mechanisms

Mechanical power take-off mechanisms allow a direct power transfer from engine to implements offering the best possible efficiency of power transfer.

Modern tractors of technology levels III, IV, and V offer a rear PTO with 2–4 speeds and mostly as an option a front PTO with usually one speed, **Fig. 5.13**. But there are also simpler configurations, **Fig. 5.14**.

Structure A is the simplest one, with no extra PTO clutch. When the tractor stops, so does the implement, which in many cases is a disadvantage. Hence, structure A is a low-cost option only for technology level I.

Structure B has a separate PTO clutch which is integrated with the ground drive master clutch in order to save costs, typically for technology levels II and III. At least two speeds are common, often also additional "eco" speeds at reduced engine rpm.

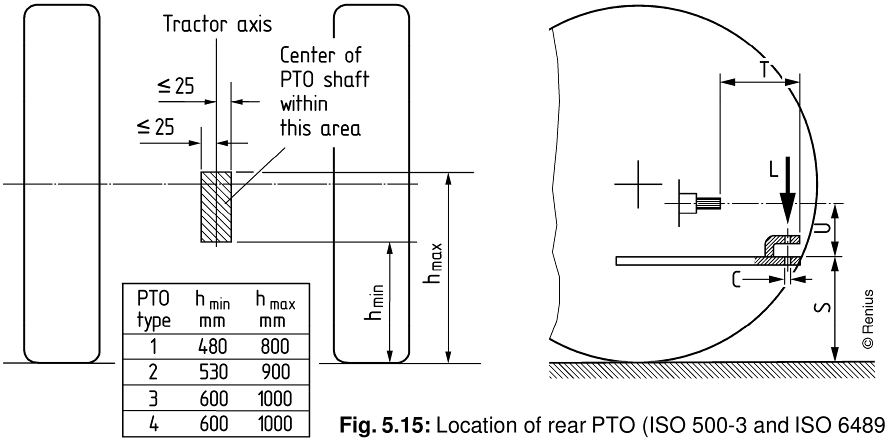
Structure C has similar functionality to that of B, but uses a separate wet multi-disc PTO clutch in the rear part. The costs are much higher but the "life" shaft enables driving of auxiliaries, e. g., hydraulic pumps, typical for technology levels IV and V.

Structure D offers, in addition to A, B, and C, a ground speed PTO; for example, applied as option to drive trailers with a synchronized axle speed. A former GDR standard cited 540 rpm at 3.3 km/h. For an actual relation see, for example, Fig. 5.85.

5.4.2 Location of rear PTO shaft

A standardized location of the rear PTO shaft is essential in order to secure a proper power transfer to tractor implements.

The life of the connecting shaft benefits from limited inclination. The dimensions of **Fig. 5.15** and **Table 5.7** are in accordance with ISO 500-3 and ISO 6489-3**.**



PTO type	h min mm	h max mm
1	480	800
2	530	900
3	600	1000
4	600	1000

Fig. 5.15: Location of rear PTO (ISO 500-3 and ISO 6489-3)

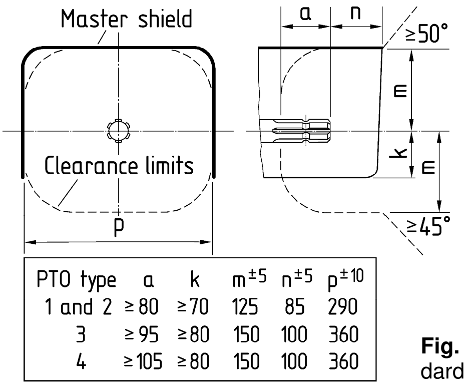
Table 5.7: Location of drawbar and rear PTO shaft as standardized in ISO 6489-3

Drawbar category	0	1	2	3	4	5
Max. static vertical load L [kN] in regular drawbar position	5	8	15	18	22	30
Nominal height S	220–420	330–500		380–560		400–600
PTO distance U	160	220	250	260	280	310
PTO distance T	about 400 for regular drawbar length					
Pin hole diam. C	20	33		41	52.5	72.5
Pin diameter	18	30		38	50	70

All dimensions in mm

A **protective guard** of the PTO drive shaft with a defined clearance is a *safety must* in many countries, **Fig. 5.16**. The dimensions are in accordance with ISO 500 for standard tractors (dimensions for narrow track tractors are different).

Safety specifications for the front PTO (ISO 8759) are similar to those of the rear PTO (see also chapter 7).



PTO type	a	k	$m^{\pm5}$	$n^{\pm5}$	$p^{\pm10}$
1 and 2	≥ 80	≥ 70	125	85	290
3	≥ 95	≥ 80	150	100	360
4	≥ 105	≥ 80	150	100	360

Fig. 5.16 (left)**:** Master shield of the rear PTO for standard tractors. Dimensions in mm as of ISO 500

5.4.3 PTO design fundamentals

The mechanical tractor power take-off and the three-point linkage are the most important components of the tractor-implement interface. Increased tractor power has led from one PTO type to now four standardized versions, **Table 5.8** and **5.9**.

Table 5.8: Basic specifications of mechanical power take-off (PTO), as standardized in ISO 500

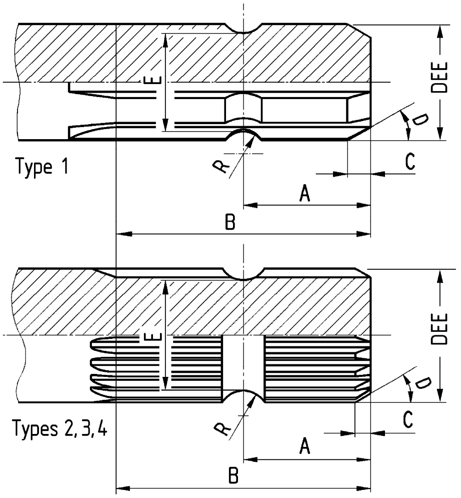
PTO type	Nom. diameter mm	Number of splines	Type of splines	Nom. rotational speed*), rpm	Max. power at the shaft, kW	Max. telescoping thrust force, kN
1	35	6	straight	540 1000**)	65 110	≤ 65 kW: ± 9 65 - 130 kW: ± 12
2	35	21	involute	1000	130	
3	45	20	involute	1000	300	±18
4	57.5	22	involute	1300	450	not defined (± 22?)

*) Engine speed is not specified in ISO 500 enabling eco-PTOs (nominal PTO speed at reduced engine speed). Popular basic equipment: Nom. PTO speeds at about 90% nom. eng. speed for type 1 and 95% for types 2-4

**) Type 1 not available with 1000 rpm for North America

Table 5.9: Dimensions for the PTO shaft as specified in Fig. 5.17 and standardized in ISO 500-3

Dimensions		Type 1	Type 2	Type 3	Type 4
DEE	Major diameter, mm	34.87 -0.12	34.874 -0.025	44.425 -0.025	57.500 -0.025
A	Groove to end of shaft, mm	38 ± 0.8	25.5 ± 0.8	38 ± 0.8	50 ± 0.8
B	Effective spline length and hardenend portion, mm	≥76	≥64	≥89	≥100
C	Chamfer, mm	6 +1	5 +1	6 +1	8 +1
D	Chamfer angle, deg	30° ±3°			
E	Inner dia. of groove, mm	29.40 ±0.1	29.40 ±0.1	37.25 ±0.1	48 ±0.1
R	Radius of groove, mm	6.8 ±0.25		8.4 ±0.25	10.4 ±0.25



Important ISO standards for the PTO are:

- ISO 500: Rear mounted PTO (3 parts)
- ISO 8759 Front mounted PTO (3 parts)
- ISO 5673-1 Drive shaft configurations, safety aspects, axial telescopic forces
- ISO 5674 Safety guards
- ISO 26322 General tractor safety (2 parts)
- ISO 4254-1 General safety

The front PTO has the same dimensions as the rear PTO with four types as well. A rotation opposite to the engine rotation is possible with only two gear wheels. However, ISO tolerates both PTO directions.

Fig. 5.17 (left): Two basic geometries of PTO shafts. Measures as standardized in ISO 500

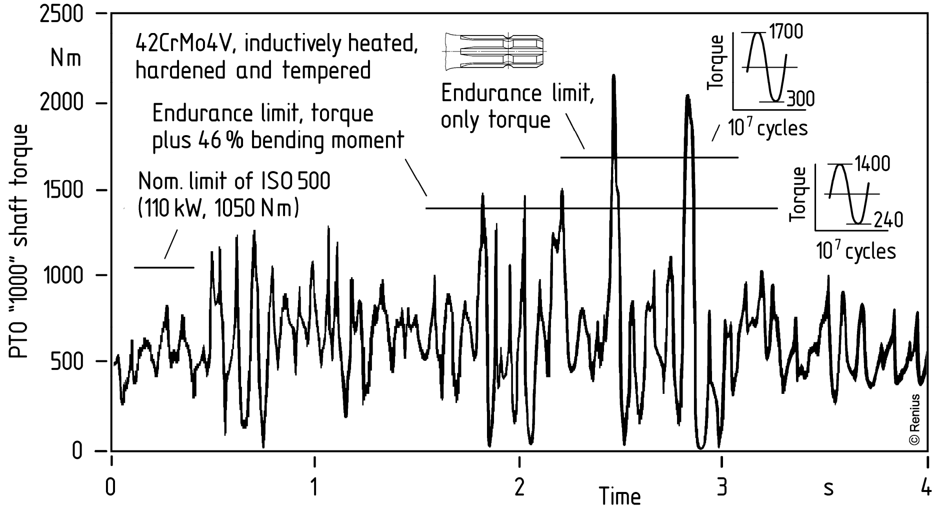


Fig. 5.18: Torque loads at the rear PTO *shaft type 1 (6 splines)* with 1000 rpm for a heavy power tiller operation on a firm meadow, width 2.5m. Nominal tractor engine power 92 kW [5.26]. Endurance limits for 42CrMo4V without additional bending according to Fig. 5.19. Superimposed bending is considered at a 17% endurance limit reduction [5.29]. All endurance limits for 50% probability

PTO shaft life. Geometry is widely fixed (ISO 500). There are three types of loads:

- Telescoping forces [5.25]
- Torque loads [5.26–5.28]
- Bending moments [5.29]

Telescoping forces are important for transmission bearings and case wall thickness but not for the shaft durability. Torque moments can be extremely cyclic; **Fig. 5.18** [5.26]. Superimposed bending reduces shaft torque limits and hence power performance. Main reasons for these dynamics are pulsating implement loads and/or inappropriate positions of the universal joints [5.27].

The PTO clutch slipping torque and additional torque limiter are common to prevent extreme torque peaks [5.30].

Simple carbon steels such as SAE 1040 F were common around 1950 [5.27], now replaced by high-performance steels such as SAE 4140 (42CrMo4V). Optimized inductive heat treating and shot peening enables high dynamic torque performance, best represented with S-N curves, scaled in torque load, **Fig. 5.19**. They enable service life forecasts with load spectra. Carburized steels such as SAE 5120 (20MnCr5) offer slightly lower performance.

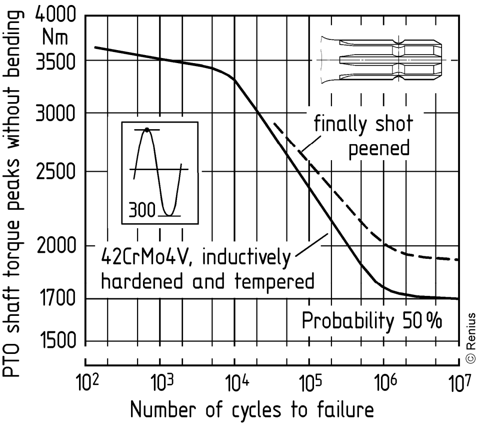


Fig. 5.19: Dynamic torque load capacity for PTO shaft type 1, without bending [5.26, 5.29, 5.30]

Additional bending moments reduce the torque capacity by about 15–20 % [5.29].

Flanged shafts simplify their change.

Torque load spectra for gear wheels and bearings of the PTO powertrain can be best picked up by speed triggered sample counting. As the speed variation is not high, time portions are usually sufficient.

Rainflow spectra cannot be used in this case as they cannot cope with local stress dynamics of gear wheels and bearings.

The highest level of PTO torque loads can be found for power tillers (Fig. 5.18) and for large square balers, **Fig. 5.20**.

These implements with highest peaks of about 400% of the average load outweigh most other torque load levels.

Service life analysis can be made by cumulative damage theory, for gear wheels by using the *Miner-Haibach* procedure [5.23]. **Fig. 5.21** is a proposal for the full life torque load distribution. Results of Fig. 5.20 are introduced with 20%. A nominal PTO torque of 1210 Nm at 1025 rpm corresponds to 130 kW nominal PTO power.

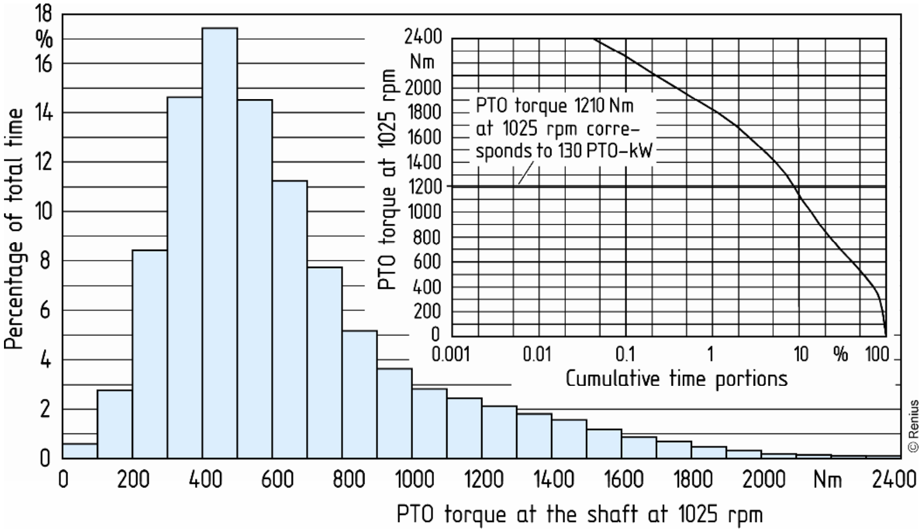


Fig. 5.20: PTO torque load spectra for driving a big pack baler: class frequency, also called histogram (left) and derived cumulative load spectrum (right). Measurements of Walterscheid [5.30]

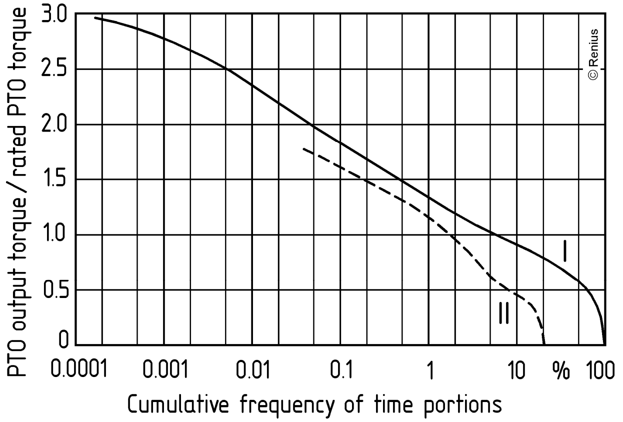


Fig. 5.21 (left)**:** PTO load spectra

I – Proposed total distribution for a typical mixture of operations, modified from [5.28]. The author thinks that I covers 90–95 % of all practical operations for an engine torque back-up of about 25 % (corresponds with about 30 % at PTO shaft)

II – Measured load spectrum of Fig. 5.20 (heavy load) for a tractor with 130 kW nominal PTO power (which may need a type 2 shaft) and 20 % on-time of total life, well covered by I

5.4.4 Examples of PTO powertrains

For a long time, the two-speed PTO was the normal version with standardized speeds of 540 and 1000 rpm at about 90–95% rated engine speed.

This relation is useful for heavy loads in order to use the full engine performance. But fuel efficiency is poor in the case of low loads/high engine rpms (Fig. 5.2).

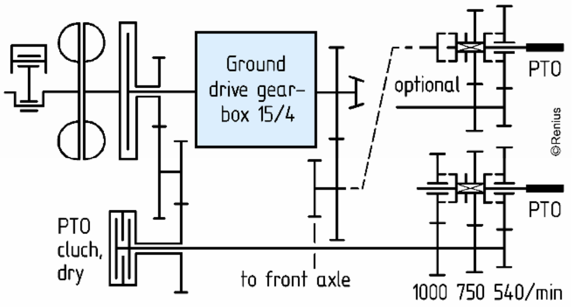


Fig. 5.22: First fuel-saving PTO speed “750 rpm” offering 540/min at reduced engine speed, Fendt 1982 [5.31]

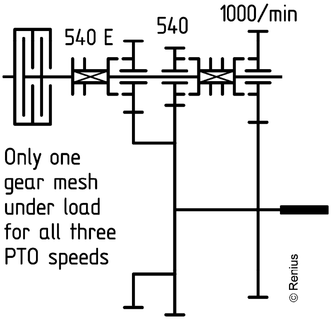
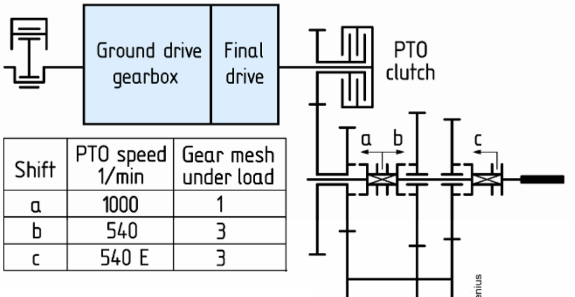


Fig. 5.23: PTO structure of CLAAS Axion 800 line (120–165 kW ECE R 24), as of 2010 [5.32]



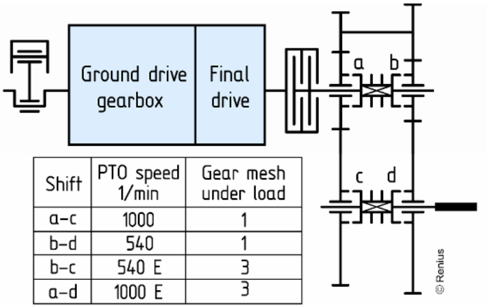
Shift	PTO speed 1/min	Gear mesh under load
a	1000	1
b	540	3
c	540 E	3

Fig. 5.24: PTO powertrain of J. Deere 6R line (92–176 kW 97/68/EG), as of 2012 [5.32]

This was the reason for developing fuel-saving PTO structures with nominal PTO speeds at lower engine speeds, **Fig 5.22**. PTO powertrains with only one gear mesh under load, **Fig. 5.23**, offer good efficiencies but need large gear wheels.

The distribution of the large speed reduction ratio (about 4:1 for 540 rpm) on more gear wheels leads to a more compact design, **Fig. 5.24** and **5.25**. As the 1000/min PTO is preferred in this power class, it has still one gear contact.

Proposals have been presented to introduce power shifted or CVT PTOs [5.32–5.34]. Problems of highly sophisticated solutions are the very limited space at the rear part of the tractor and the extremely high PTO efficiency demand (PTO power is the price basis in some countries).



Shift	PTO speed 1/min	Gear mesh under load
a–c	1000	1
b–d	540	1
b–c	540 E	3
a–d	1000 E	3

Fig. 5.25: PTO powertrain of Case IH PUMA line (120–165 kW ECE R 24, 2013), very compact in length, needs a special shift control. Priority for nominal PTO speeds 540 and 1000/min [5.32]

A PTO structure of a ZF transmission with four speeds is shown in Fig. 5.39 below.

A compact power split PTO is considered in [5.34] along with the vision of “introducing more electricity” for hybrid tractor drives.

5.5 Examples of important stepped transmissions

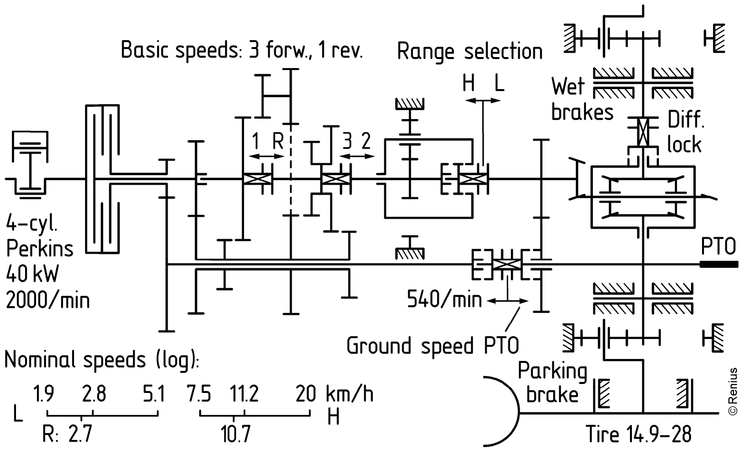


Fig. 5.26: Level I: transmission 6/2 of MF 65, 1958. Since 1962 also 12/4 with "Multi Power"

The MF transmission of **Fig. 5.26** is an example of tech level I. Six speeds forward and two reverse are realized by three sliding gears and a standard planetary set for two ranges. The rear axle with wet brakes and planetary final drives was very advanced at that time (1958). The "Multi Power" (Fig. 5.5) could double the speeds to 12/4.

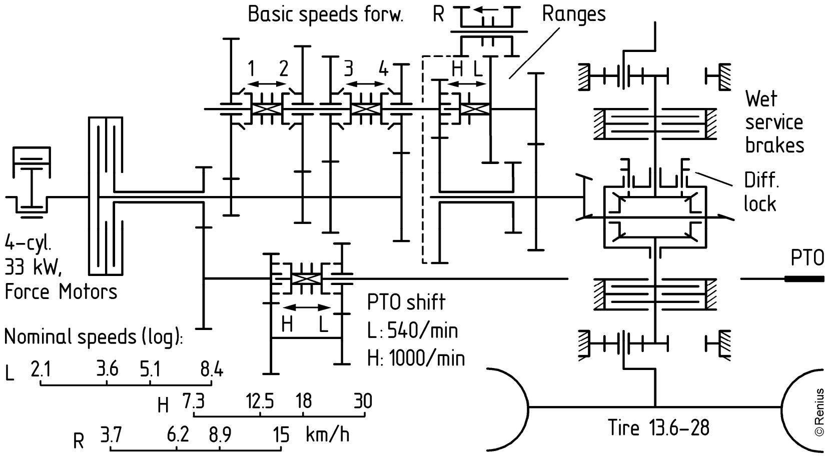


Fig. 5.27: Level II: transmission 8/4 of FORCE Motors/India (1997), based on a technology transfer using a licence for the ZF A 212 transmission (modified) and German consultation support [5.35]

The level II transmission of **Fig. 5.27** was realized in India by FORCE Motors (formerly "Bajaj Tempo") on the basis of a licence for the ZF transmission A 212.

The author supported modifications for the Indian market and defined a load spectra test program "India" [5.35]. This transmission was still competitive in 2019.

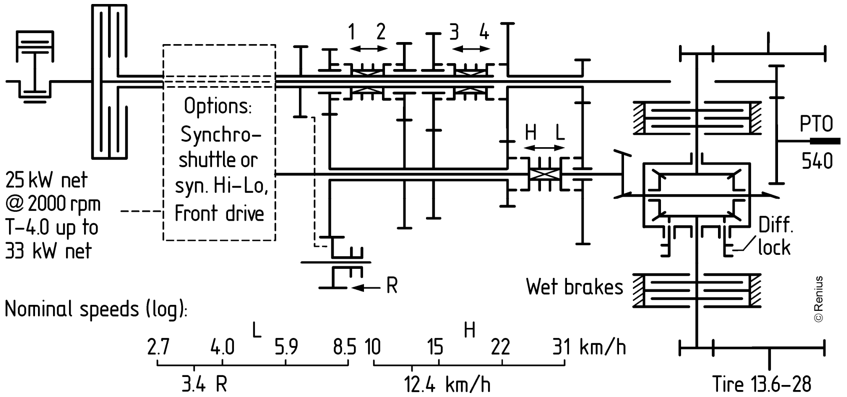


Fig. 5.28: Level II–III transmission system from CARRARO for CNH/India, Version T3-0 "basic", 2011 (Level II). Only 10 gear wheels forward and two for reverse for 8/2 speeds being very cost-saving (Fig. 5.11 and Table 5.6). The modular system allows for a wide variety of versions [5.20]

The CARRARO transmission of **Fig. 5.28** and **Fig. 5.29** represents a modular design with an extremely wide range of versions and options meeting technology levels II and III. The gearbox uses the former Deutz configuration (Fig. 5.12); however, not integrating the reverse speeds in the ranges and with no speed overlapping of the two forward ranges. However, this structure of geometrical speed spacing of the basic gears favours the speed pattern in the case of additional Hi-Lo splitting ranges (16/4).

Three versions are offered for the PTO:

1 Basic version, not independent, Fig. 5.29
2 Version "independent" (Fig. 5.28)
3 Version "independent with two speeds"

Transaxle version T4 offers planetary final drives instead of bull gears (Fig. 5.28).

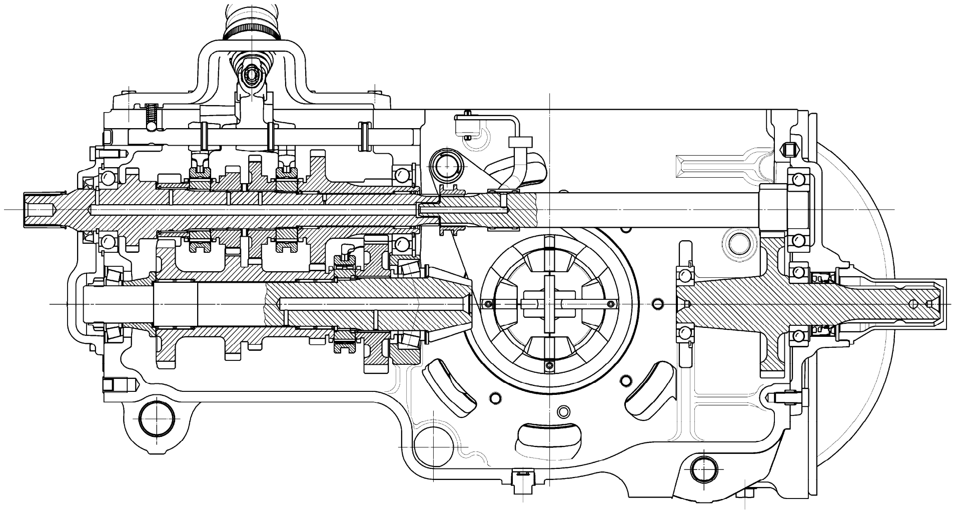


Fig. 5.29: CARRARO transmission T3-0 "basic" in the very simple low-cost version with collar shift and transmission driven PTO (stops together with ground drive), 2011. Courtesy Carraro Drivetech

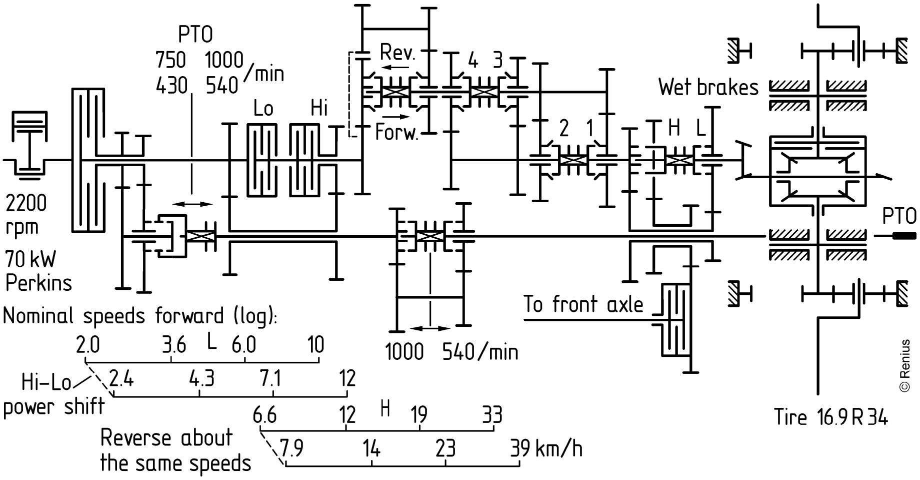


Fig. 5.30: Level III: ZF transmission family T517/T537/T557 for power inputs of 47–70 kW, several versions. This 16/16 example of T 557 is adapted to a tractor of Lindner/Austria, as of 2014

Many transmissions at technology level III could be named. ZF offers the T 500 series originating from Steyr, **Fig. 5.30**. Sixteen forward and reverse speeds (synchronized) result from four synchronized basic gears, a Hi-Lo power shift and constant mesh H-L. Creeper speeds, two economy PTO speeds, and a ground speed PTO (not shown) are available as options.

Semi power shift transmissions with four or more narrowly spaced power shifted speeds achieved a general breakthrough with the Case IH Maxxum 1989 [5.36], **Fig. 5.31**. The central master clutch decouples the power shift mass inertia from the synchro shift, limiting synchro shift dimensions and costs. 40 km/h versions are realized by adaption gear wheels increasing all speeds.

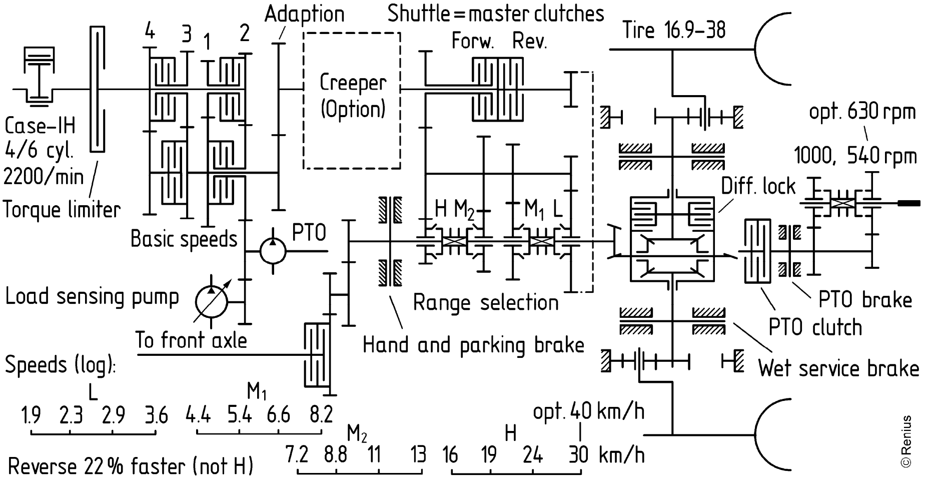


Fig. 5.31: Level IV: Case IH transmission 16/12 with four narrowly spaced power shifted speeds and power reverser ("shuttle"), introduced in 1989 for the new "Maxxum" line (66/74/81 kW) [5.36]

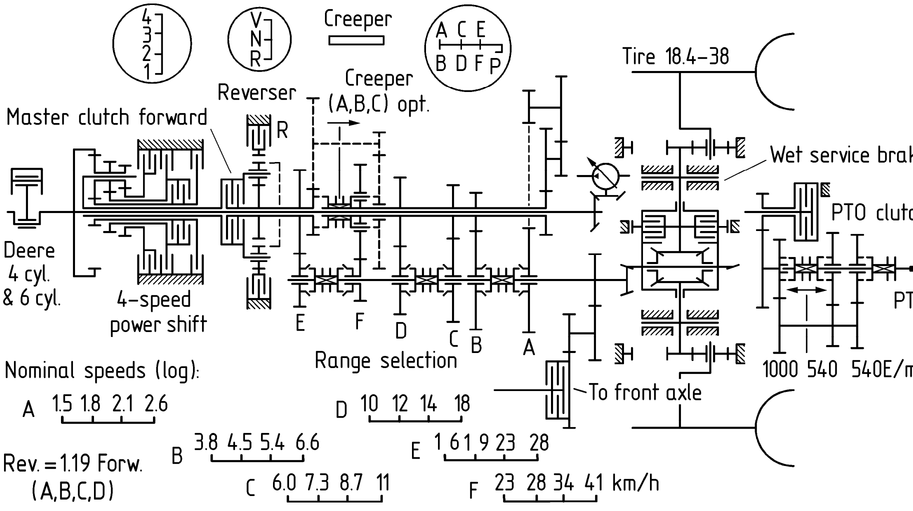


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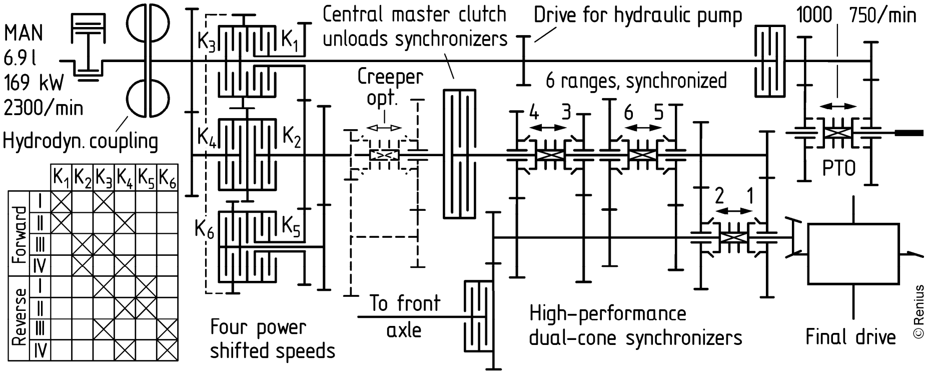
Fig. 5.32: Level IV: John Deere PowrQuad and AutoQuad 24/16 transmission with 4 power shifted narrowly spaced speeds, six ranges, and power reverser, original status as introduced in 1992 [5.37]

John Deere introduced its PowrQuad in 1992 together with the new 6000 and 7000 tractor families, **Fig. 5.32** [5.37]. The left planetary unit has three brakes and only one clutch, simplifying the hydraulic control (only one unit with oil transition into the rotating shaft). The space-saving power reverser is also applied in other John Deere transmissions. The PowrQuad was later upgraded with automatic shift strategies achieving a very high production volume.

The ZF transmission family T 7000 was presented in 1993 [5.38]; the T 7200 with three and the T 7300 with four power shifted speeds. The first customer was Fendt in 1993, **Fig. 5.33**, later other important customers. ZF improved the T 7000 family over the years by a 4-speed PTO and many automatic functions, including the synchro shift of the six ranges. Since 2016, an advanced family “TerraPower” offers six power shifted speeds [5.39], Fig 5.39 below.



		K_1	K_2	K_3	K_4	K_5	K_6
Forward	I	☒		☒			
	II	☒			☒		
	III		☒	☒			
	IV		☒		☒		
Reverse	I			☒		☒	
	II				☒	☒	
	III			☒			☒
	IV				☒		☒

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Fig. 5.33: Level IV: Transmission of the Fendt 824 (169 kW) with the initially ZF T 7000 version, the T-7300. Typical for Fendt was the hydrodynamic coupling. A complete ZF transaxle [5.38] was used from 1995 by Deutz-Fahr and Case-Steyr. The rear PTO was later entended to four speeds

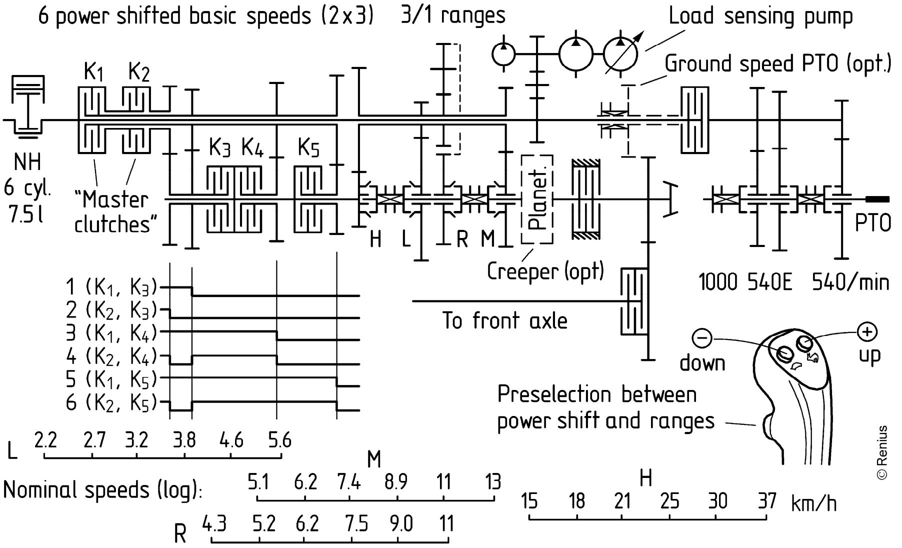
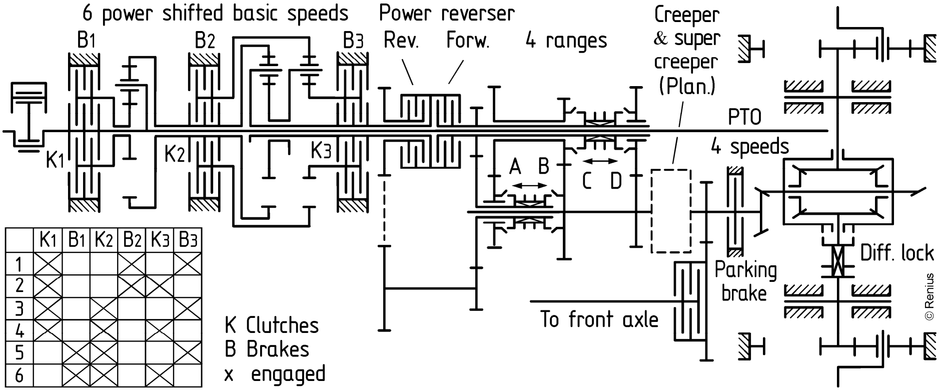


Fig. 5.34: Level IV: New Holland "Range Command" 18/6 for Fiat and Ford 1996, 74–119 kW [5.40]

New Holland presented its Range Command in 1996 (USA) for Ford and Fiat tractors, **Fig. 5.34** [5.40]. Six narrowly spaced power shifted speeds forward (with only four pairs of gear wheels) are combined with three forward ranges and one reverse. Synchronized reversing is supported in the retardation phase by engaging both master clutches K1 and K2 at the same time, resulting in a braking function. All shifting elements are actuated electro-hydraulically.

MF presented its four-speed power shift Dyna 4 (with 8 ranges and reverser) at SIMA 1992 for the MF 3600 line. GIMA (F) was founded in 1994 by Renault and MF, completing the power shift section in 2005 by a third planetary set offering now six power shifted working speeds in four ranges forward with a central power reverser and optional creeper speeds, **Fig. 5.35**. Although competitors, MF and CLAAS enjoyed great success with this transmission.



	K1	B1	K2	B2	K3	B3
1	☒			☒		☒
2	☒			☒	☒	
3	☒		☒			☒
4	☒		☒		☒	
5		☒	☒			☒
6		☒	☒		☒	

K Clutches
B Brakes
x engaged

Fig. 5.35: Level IV: GIMA 6-speed power shift (2005), versions "Hexashift" for CLAAS and "Dyna 6" for MF. This transmission map represents the structure for CLAAS Arion and Axion, as of 2014

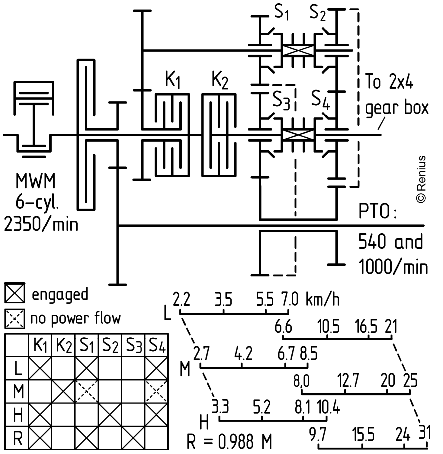
The dual-clutch principle enables realizing more than two power shifted speeds with only two clutches.

It was invented as early as 1939/40, but used commercially much later, **Table 5.10**.

The initial idea was a 1939 patent of the automobile engineer *Adolphe Kégresse* and half a year later a patent of the German agricultural engineer *Rudolf Franke*. They did not meet nor know one another as outlined in [5.41]. Franke *could even not know* of Kégresse's patent, as it was initially not published in Germany. Neither during my various communications with him nor in his own publications such as in ATZ [5.42] did he mention the name *Kégresse*,.

In 1965, IHC presented a 4-speed dual-clutch range structure [5.7] and ZF in 1984 the T6500 with 8 power shifted dual-clutch speeds [5.44] for Schlüter in a limited production volume until 1989.

Many patents on the subject have been filed [5.41]. In 1991, Renault presented the "Tracto-Shift", a three-speed power shift with power reverser and eight ranges [5.42] using the dual clutch principle, **Fig. 5.36**.



☒ engaged
⌧ no power flow

	K1	K2	S1	S2	S3	S4
L	☒		☒			☒
M		☒	⌧			⌧
H	☒			☒		☒
R	☒		☒		☒	

Fig. 5.36: Renault TRACTO-SHIFT 24/8 transmission with a 3/1 power shift unit using the dual-clutch principle. Presented at the 1991 SIMA Paris for series 54 (74–107 kW) [5.43]

Table 5.10: Dual-clutch principle, milestones

1939	*Adolphe Kégresse* (1879-1943), Patent DBP 894 204 (12.08.1939)
1940	Prof. *Rudolf Franke* (1906-2001) Patent DBP 923 402 (20.03.1940) et al.
1965	IHC tractor 523: 4 ranges with dual clutch
1984	Dual clutch transmissions in Porsche racing cars, positive experience
1984	ZF tractor transmission T 6500 8-speed power shift with wet dual clutches, no breakthrough
1989	Proposals of *R. Franke* in ATZ Vol. 91 (1989)
1991	Renault 3-speed power shift TRACTO-SHIFT with power reverser
1999	Landini 3-speed power shift DELTASIX with power reverser
2003	Start of large series production of dual clutch passenger car transmissions at VW, Kassel
2004	Ph.D. Thesis of *Thomas Götze* (Karlsruhe)
2008	Dual cluch gear boxes by GETRAC (mainly for BMW), ZF (for Porsche 911, etc.)
2009	First application of the principle for a tractor power split CVT (ranges) by CNH Industrial
2010	Honda VFR 1200 F, motor cycle with dual clutch transmission
2012	John Deere "DirectDrive" with 8-speed power shift, using dual clutch principle
2017	CNH 8-speed power shift

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A further application is in the tractor transmission DELTASIX of Landini (1999) [5.45] for the "Legend" line, **Fig. 5.37**, offering three power shifted speeds forward and a power shuttle, combined with up to 18 conventional basic speeds. Power shift is possible for H-M, L-M, and R-M.

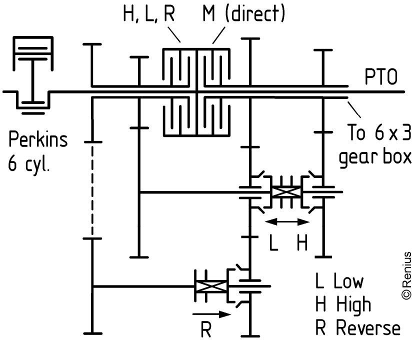


Fig. 5.37: Landini DELTASIX 54/18 transmission with a 3/1 power shift using the dual-clutch principle. Presented in 1999 for the "Legend" line (81–129 kW) [5.45]

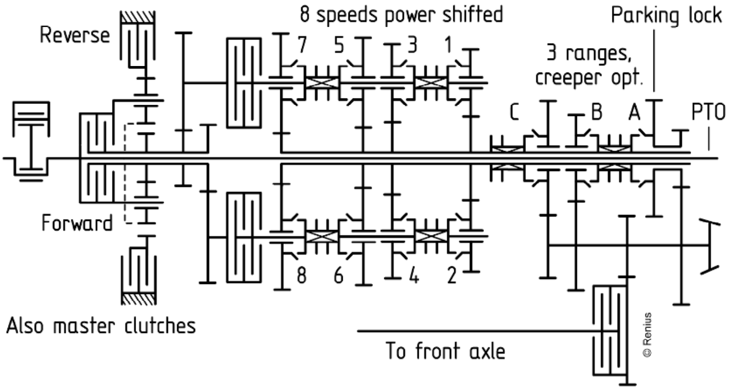
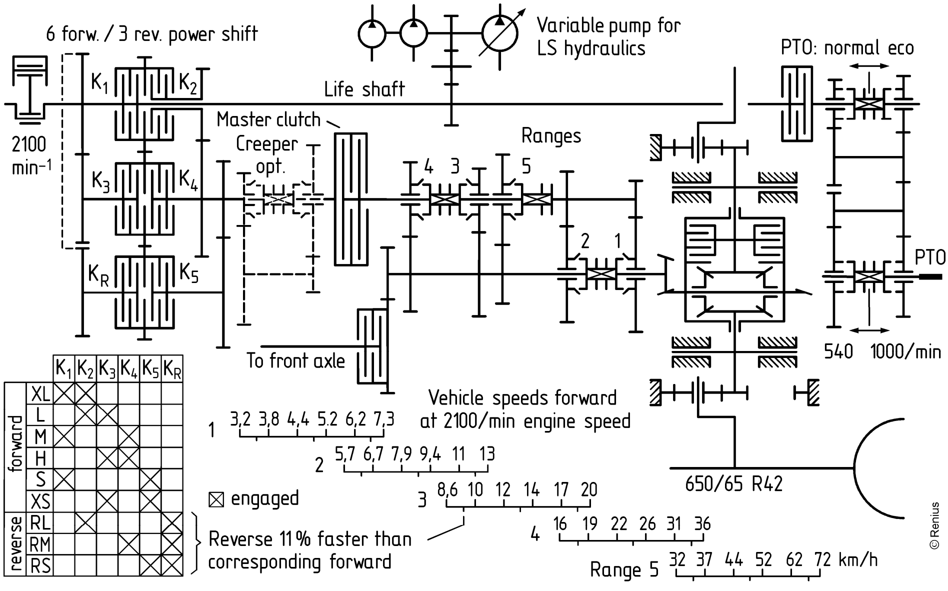


Fig. 5.38: John Deere 24/24 “DirectDrive”: Eight power shifted speeds using the dual-clutch principle with power reverser and three ranges. Introduction in 2012 for tractor series 6R [5.46]

In 2012, John Deere introduced the 24/24 “DirectDrive” with an 8-speed dual-clutch power shift for its 6R line, **Fig. 5.38** [5.46], power shifting between even and odd speeds. The design needs only two clutches but as many as 16 gear wheels. For the first time in a tractor, electromagnets are used for synchronized shifts, saving control energy. The power shuttle is a well-known design.

In 2016, ZF introduced the transmission family TPT 16 and TPT 20, **Fig. 5.39**, now offering 6/3 power shifted speeds (the former T 7200/7300 [5.38] had 4/4). The speed step ratios were again reduced to now only about 1.18, although one range was saved from previously six and no clutch has been added. The first customer was Deutz-Fahr in 2016 for its tractor line 6.



		K_1	K_2	K_3	K_4	K_5	K_R
forward	XL	☒	☒				
	L		☒	☒			
	M	☒			☒		
	H			☒	☒		
	S	☒				☒	
	XS			☒		☒	
reverse	RL		☒				☒
	RM				☒		☒
	RS					☒	☒

☒ engaged

Reverse 11 % faster than corresponding forward

Fig. 5.39: ZF transmission family TPT 16 and TPT 20 (2016) with about 150 kW max. net input power, 6/3 power shifted speeds forward/reverse. Four or five ranges (5 offering 40/50/60 km/h at reduced diesel engine speeds for the top speeds, saving fuel). Data courtesy ZF

Full power shift tractor transmissions allow *changing all speeds under power* without any operation of a clutch pedal, similarly to automatic car transmissions.

However, the number of speeds in tractor transmissions is usually much higher than in other vehicles. Changing speeds necessitates a highly sophisticated control system, which considers load and speed levels being very different on-road and off-road.

Tractor transmissions of this type have been developed mainly in the USA with the following milestones:

- 1958 Ford "Select-O-Speed" 10/2 [5.10]
- 1964 J. Deere Power Shift 8/4 [5.11]
- 1982 Deere Power Shift 15/4 [5.18]
- 1987 Case-IH Power Shift 18/2 [5.19]
- 1990 Ford-Funk Power Shift 18/9 [5.36]
- 1994 J. Deere New Power Shift 16/5 [5.47]
- 2000 NH PowerCommand 18/6 [5.48]

The transmission map of the first design, the Ford "Select-O-Speed", was already presented and commented on in chapter 5.1. 10 speeds were a lot at that time, although gears 2 and 3 had about the same speed.

The first structures of Ford and John Deere used planetary sets while most later designs preferred countershaft structures.

In 1994, John Deere introduced a completely new full power shift transmission for the new large 8000 series, **Fig. 5.40**. The 4 x 4 structure offers a fine concentration of speeds within the main working range due to the overlapping of ranges B and C.

Reverse speeds 8.9 and 10.8 km/h are represented by range D with reduced engine speeds while top speed 38 km/h is achieved by an engine speed above 2200 rpm. The high offset of engine and rear axle results from the high engine position.

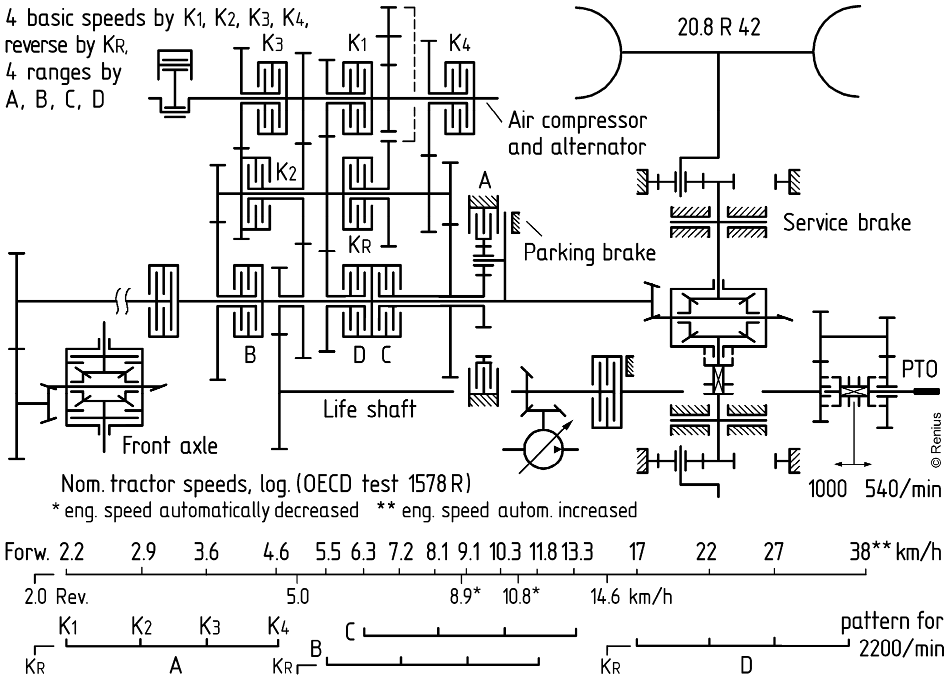

4 basic speeds by K1, K2, K3, K4,
reverse by KR,
4 ranges by
A, B, C, D
K3
K1
K4
K2
KR
A
B
D C
20.8 R 42
Air compressor and alternator
Service brake
Parking brake
Life shaft
Front axle
PTO
1000 540/min
© Renius
Nom. tractor speeds, log. (OECD test 1578 R)
* eng. speed automatically decreased ** eng. speed autom. increased
Forw. 2.2 2.9 3.6 4.6 5.5 6.3 7.2 8.1 9.1 10.3 11.8 13.3 17 22 27 38** km/h
2.0 Rev. 5.0 8.9* 10.8* 14.6 km/h
K1 K2 K3 K4
KR A
C
B
KR
KR D
pattern for 2200/min

Fig. 5.40: John Deere 16/5 full power shift for the new 8000 line (136–191 kW) 1994 [5.47]. The speeds are taken from a diagram of OECD Test No. 1578 R. Fine speed concentration but top speed 38 km/h by increased engine speed and no power shuttle initiated a follower (Fig. 5.42)

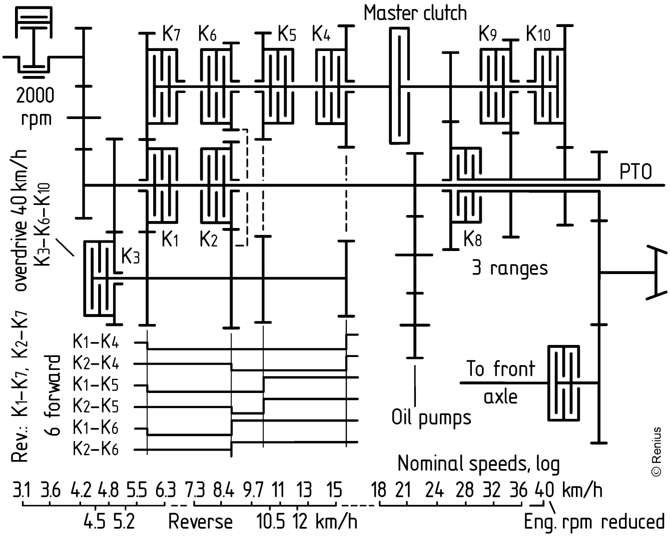


Fig. 5.41: 19/4 full power shift from CNH (Case IH Magnum and NH T 8000, 2012) [5.48]

In 2012, CNH Industrial introduced its new 19/4 full power shift for large Case IH and New Holland tractors, **Fig. 5.41**. The overdrive is optional, with a reduced engine speed at 40 km/h. The basic structure is still slightly similar to that of 1987 [5.19].

John Deere introduced in 2014 the new 23/11 power shift e23, **Fig. 5.42**, now offering geometric speed spacing, improved reverse speeds, and top speeds at reduced engine speeds. One of 24 speeds remains unused for better shifting ability [5.49].

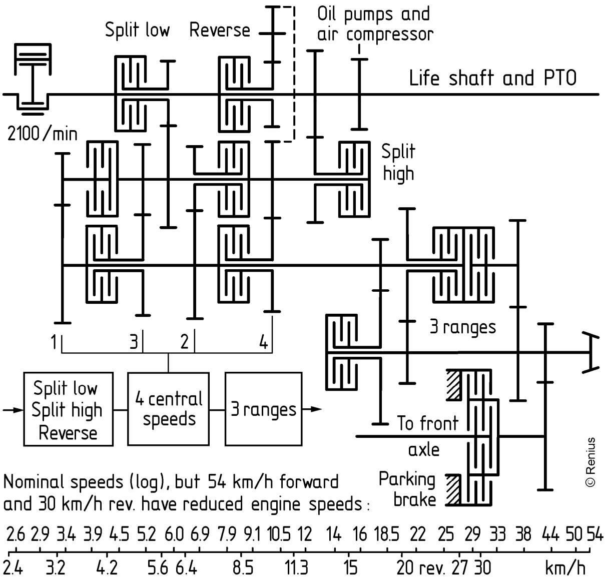


Fig. 5.42: John Deere e23 (23/11) full power shift for 7R series (2014), later also 8R series [5.49]

5.6 Continuously variable transmissions (CVTs)

5.6.1 CVT principles – a survey

Continuously variable tractor transmissions are considered to be the ideal technology offering a vehicle speed which is independent of the engine speed (Fig. 5.2).

A summary of their history (chapter 5.1) has demonstrated that much money has been spent for research and development with many disappointments and even flops.

Principles of CVTs are listed in **Table 5.11**.

Hydrodynamic torque converters are the cheapest CVTs representing the by far highest production volume of all CVT types (passenger cars!) but they have two major weak points regarding their use in tractors:

- The maximum efficiency is only available within a very narrow band of ratios.
- The ratio cannot be controlled in a closed loop as it is self-adjusting.

The first weak point can be diminished by a freewheeling element or a clutch blocking the unit. But this reduces the CVT effect; thus requires a high number of additional ranges. A solution to diminish the second weak point is to apply adjustable guiding vanes. Hydrodynamic CVTs were not introduced for mass-produced tractors.

Mechanical CVTs using chain drives can achieve *the best efficiencies of all known physical principles if the hydraulic auxiliary system is optimized for a minimum of its own power consumption* [5.51].

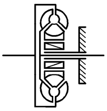
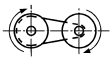
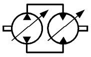
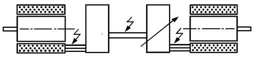
Both pull type and push type versions have been applied in series for vehicles.

Toroidal CVTs need higher slip values and offer slightly lower efficiencies than push type chain versions [5.50].

Hydrostatic "direct" drives became popular for small tractors – early in Japan (with high exports) and USA. Solutions for larger standard tractors failed, mainly because of only moderate efficiencies.

Electrical CVTs are under research with prototypes. The efficiency of diesel-electric CVTs has not been impressive so far. Electric power split CVTs might have slightly better efficiencies than present hydrostatic power split structures do; one reason John Deere announced a development in 2019.

Table 5.11: Physical principles of "direct" continuously variable transmissions for vehicles [5.50]

Type of CVT	Principle of energy transmission	Ratio control	Typical application	Efficiency
1 Hydrodynamic 	Fluid mass forces at pump and turbine	Usually automatically by load	Passenger cars and construction machinery	poor
2 Mechanical 	Traction forces of friction contacts	Radius of traction force	Passenger cars, tractors	excellent
3 Hydrostatic 	Hydrostatic forces at pump(s) and motor(s)	Displacement of the units	Mobile machinery	moderate
4 Electrical 	Electromagnetic forces at generator and motor(s)	Frequency of current	Passenger cars, forklift trucks	good

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5.6.2 Pull type steel belt CVTs

Pull type steel belt CVTs, Fig. 5.43, work in principle like an adjustable V-belt drive. Early fundamentals were presented by *Dittrich* in 1953 [5.51]. A new generation of P.I.V. CVTs became the heart of the "Munich Research Tractor" (1988) [5.50, 5.52]. Activities at TU Munich were able to improve the general fundamentals considerably, based on modelling and validation by experiments [5.53, 5.54].

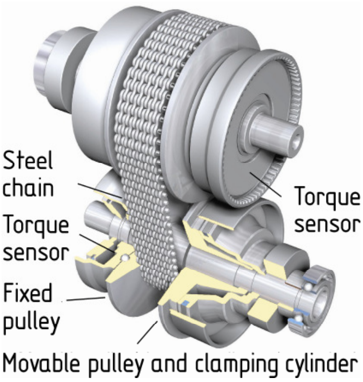

Fig. 5.43: Pull type chain CVT, P.I.V. - LuK. Actuating pressure 20–50 bar. Courtesy PIV

The P.I.V. principle did not result in series production for tractors at that time but was introduced by Audi-LuK with the "Multitronic" in 1999 for the Audi A6 [5.55]. The maximum input torque was initially 280 Nm, later 380-400 Nm, enabling power levels of 140 kW for diesel and 195 kW for gasoline engines with spreads of up to the very high value of 6.7.

Load spectra at low speeds are relevant for dimensioning. They are more bulgy for tractors than for passenger cars, allowing a lower CVT power level not exceeding 100–120 kW combined with a spread of about 5.

Fig. 5.44 demonstrates the control system. **Outstanding efficiencies** can be obtained with the following design principles:

1. Steel chain without internal sliding, only rolling at the chain pin pairs
2. Axial clamping forces automatically adjusted to the torque load at slip limits
3. Minimized auxiliary power of the hydraulic control system [5.56, 5.57]

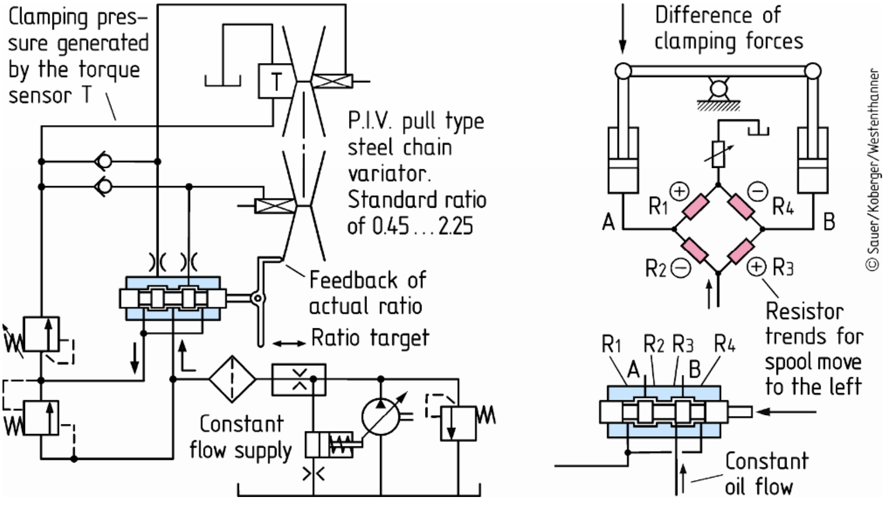
Clamping pressure generated by the torque sensor T
T
P.I.V. pull type steel chain variator. Standard ratio of 0.45…2.25
Feedback of actual ratio
Ratio target
Constant flow supply
Difference of clamping forces
A
B
R1
R4
R2
R3
Resistor trends for spool move to the left
R1
R2
R3
R4
A
B
Constant oil flow
© Sauer/Koberger/Westenthanner

Fig. 5.44: Control system of a modern pull type chain variator (left). The clamping preload is generated by the torque sensor and the ratio is controlled by the central spool valve which represents four fluid resistors in a "full" Wheatstone bridge offering very high sensitivity [5.53, 5.56, 5.57].

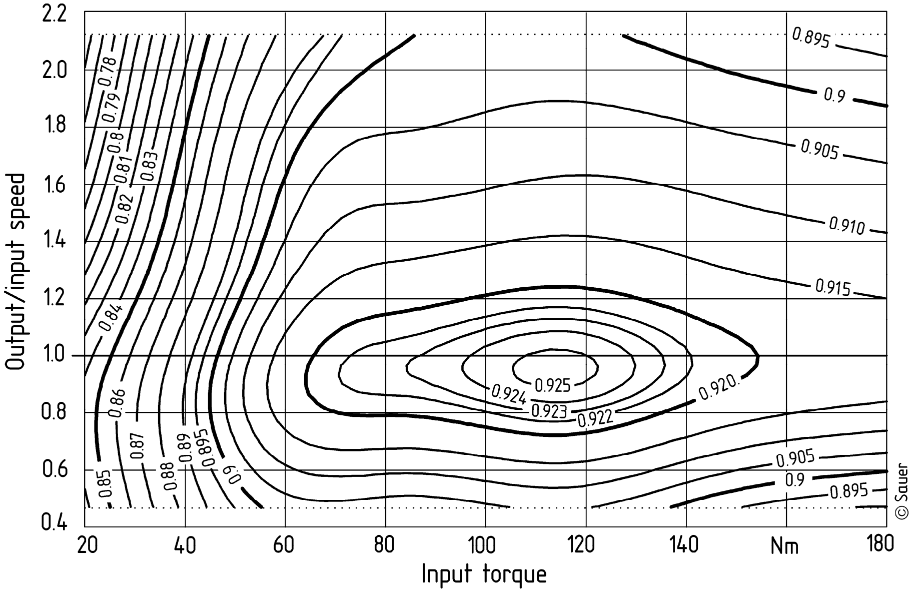

Fig. 5.45: Measured efficiencies of a P.I.V. CVT. Optimized clamping forces (torque sensors at input and output), 7 l/min constant flow supply by a variable displacement pump. Input speed 1000/min, higher values of up to about 3000 rpm input do not downgrade this level [5.53]

Research at TU Munich resulted in design fundamentals for pull type CVTs.

Optimized efficiency is in a wide range above 90%, **Fig. 5.45** [5.53] as confirmed by LuK-Audi in [5.55] and other LuK publications. High rotational speeds do not reduce the level; however, typical fatigue limits result from centrifugal chain forces at output speeds above about 10,000 rpm.

The Audi-LuK production volume was, for example, 1000 units a day in 2009 [5.55].

CVTs with friction contacts of rolling elements such as toroidal units [5.50] are slightly lower in efficiency than the level of Fig. 5.45 – one reason is the larger slip. However, they offer the possibility of higher power levels with parallel units.

Electrical CVTs have slightly lower efficiencies than those of Fig. 5.45.

This confirms the outstanding efficiency potential of pull type steel belt CVTs.

Friction drive CVTs cannot offer a zero output speed directly. However, this is possible by the "geared neutral" principle [5.58].

A prototype at TU Munich had a structure with two ranges [5.59]: a first one with power split for extending the speed spread and a second with direct power transfer. In its "Boomer" line, New Holland introduced a similar structure but with only one range, enlarged by power split, **Fig. 5.46**.

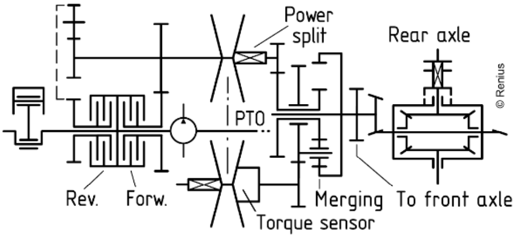

Fig. 5.46: CVT "EasyDrive" of NH "Boomer" tractor line (2009), in 2015 up to 40 kW. P.I.V. variator with extended speed spread by power split [5.60]

5.6.3 Hydrostatic CVTs

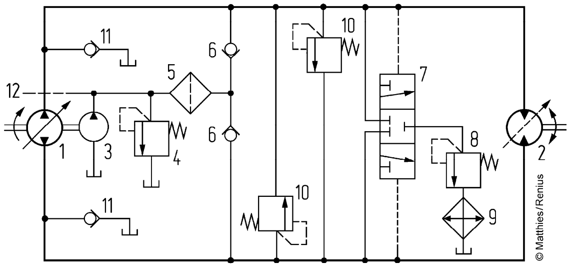

Fig. 5.47: Typical circuit diagram of a hydrostatic CVT. Reference numbers are explained in the text [5.61]

Hydrostatic CVTs (also called HSTs) are applied directly or with power split. This first chapter addresses only the elementary "direct" dives. They have been applied for small tractors since many decades in high volumes, [5.4], mostly combined with mechanical ranges. At the same time, they are the basis for modern hydromechanical power split CVTs.

The typical HST structure can best be shown by using symbols from ISO 1219, **Fig. 5.47**.

The variable displacement pump (1) works on the hydraulic motor (2), producing the CVT effect. The charge pump (3) with safety valve (4) charges the low pressure side via the check valves (6). The surplus is automatically discharged by the flushing valve (7, controlled by high-pressure side), its pressure level controlled by the relief valve (8). Heat is removed by cooler (9). The pressure relief valves (10) limit the top pressure which can, however, be realized better (throttle-free) by the pump displacement (pressure sensor). The check valves (11) deal with suction in an emergency case. The charge pressure (20–25 bar) is also used for auxiliary power of the pump and motor displacement controls (12).

The best efficiency of a hydrostatic "direct" CVT is usually only in the range of 78–84%. Values above 80% require realization of some guidelines as listed in **Table 5.12**. The charge pump circuit has five major duties which are important for the circuit, **Table 5.13**.

The most used pump and motor units are shown in **Fig. 5.48**. Swash plate units are cheap and popular as pumps. Bent axis units offer higher efficiencies, **Fig. 5.49**. They are recommended for motor(s) as they also offer low starting torque losses.

Table 5.12: Achieving high HST efficiencies

- Displacement adjustable for pump and motor
- Large angle bent axis units (45°) at least for the hydraulic motor(s)
- Motor(s) larger than the pump
- If possible, more than one motor parallel in the circuit (disconnecting one at top speed)
- Charge pump with variable displacement for flow on demand, mainly for larger CVTs
- Moderate input speeds
- Moderate pressure levels for the main operations, e. g. 250–300 bar (25–30 MPa)
- Compact design, short pipes with large diameters, flow computer simulation (CFD)
- "Dry" cases, to prevent plashing
- No hoses to limit pressure losses

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Table 5.13: The 5 duties of the charge pump

- Replace the *leaked oil* within the system
- Allow bypass *filtering* of the oil
- Remove the *heat* from the closed circuit
- *Charge the low pressure side* of the pump and the motor by prepressure for smoother running and/or higher possible unit speeds
- Deliver *auxiliary power* for remotely controlling the angles of the unit(s)

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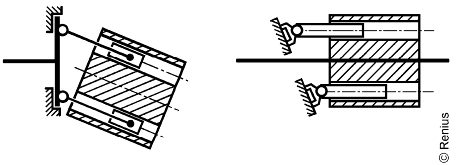

Bent axis unit, high efficiency

Swash plate unit, low initial costs

Fig. 5.48 (left)**:** Bent axis and swash plate axial piston units for hydrostatic transmissions [5.61]

Some basic equations address the physical balance between mechanical and hydrostatic power [5.61].

If any energy losses are neglected, mechanical rotational power P is equivalent to hydrostatic power by equation

$$P = T \cdot \omega = \dot{V} \cdot \Delta p \tag{5.4}$$

where

P = mechanical power (at shaft)
T = shaft torque
ω = angular velocity
$\dot{V}$ = fluid flow
Δp = pressure difference

Hydrostatic power calculation can be simplified with established "engineering units":

$$P\,[\text{kW}] = \frac{\dot{V}\,[\text{l/min}] \cdot \Delta p\,[\text{bar}]}{600} \tag{5.5}$$

The torque T is proportional to the pressure difference Δp:

$$T = \frac{\Delta p \cdot V}{2\pi} \tag{5.6}$$

where V represents the displacement of the unit in volume for one rotation.

If losses are included, the driving torque T of a pump is

$$T = \frac{\Delta p \cdot V}{2\pi} \cdot \frac{1}{\eta_{\text{mech}}} \tag{5.7}$$

where η_{mech} represents the mechanical efficiency addressing all friction and pressure losses.

Consequently, the drive of a hydrostatic motor with the torque T gives the equation

$$T = \frac{\Delta p \cdot V}{2\pi} \cdot \eta_{\text{mech}} \tag{5.8}$$

The equations (5.7) and (5.8) demonstrate that the relation between torque and pressure is not influenced by leakage.

However, the flow of a hydro pump is slightly reduced by leakage:

$$\dot{V} = n \cdot V \cdot \eta_{\text{vol}} \tag{5.9}$$

where η_{vol} represents the volumetric efficiency and n the rotational speed of input shaft.

A hydro motor runs slightly lower due to its leakage; its rotational speed n is:

$$n = \frac{\dot{V}}{V} \cdot \eta_{\text{vol}} \tag{5.10}$$

Regarding a hydrostatic transmission, the leakage is usually limited to the pump and the motor: That's why the two equations (5.9) and (5.10) usually express the total "speed losses".

The "torque losses" result from equations (5.7) and (5.8) but there are additional portions due to the losses in the pipes between the units.

The total efficiency η_{total} of a hydro pump or a hydro motor is

$$\eta_{\text{total}} = \eta_{\text{mech}} \cdot \eta_{\text{vol}} \tag{5.11}$$

Modern 45° bent axis axial piston units achieve the highest efficiencies, **Fig. 5.49** [5.61]. Swash plate units are at some percent below with major differences at low rotational speeds due to piston friction [5.62].

Further power losses must be considered for the total transmission efficiency due to the auxiliary power of the charge pump.

A constant displacement charge pump is rated for the maximum possible leakage, which is about 8–12% of the maximum flow. A variable displacement pump saves energy by flow on demand (Table 5.12).

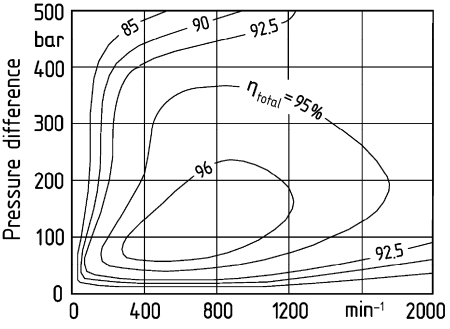

Fig. 5.49: Efficiency map of a large 45° bent axis axial piston unit. Courtesy Danfoss

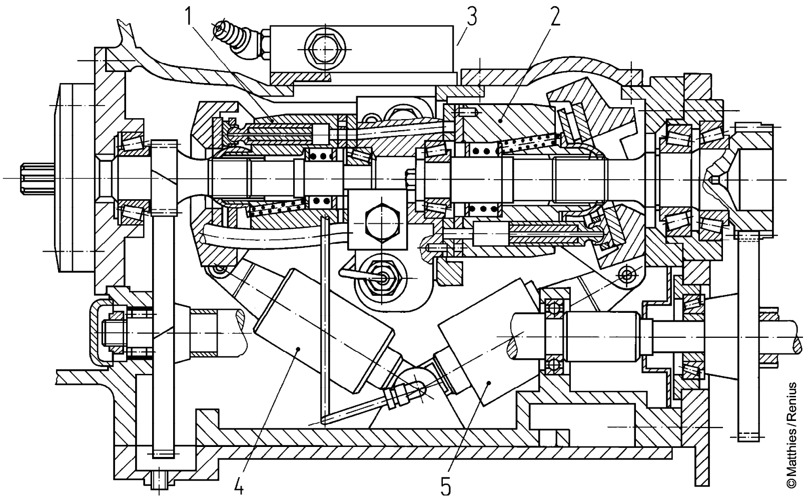

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Fig. 5.50: The first hydrostatic transmission which gained importance for professional agricultural standard tractors by series production, starting at IH with model 656 (USA 1967). Variable displacement swash plate units 1 and 2 are controlled by the valve block 3 and the servo cylinder 4 and 5. The drawing has been derived from [5.64], complete powertrain see [5.4]

Hydrostatic tractor CVTs were first applied for lawn and garden tractors [5.63]. A simplified design and a high production volume allowed for low prices.

The first application for a professional standard tractor took place in 1966 by Eicher in Germany incorporating a Dowty "Taurodyne" CVT with coupled bent axis units [5.4]. In 1972, the production was terminated at 71 units under MF.

International Harvester (IH) launched its model 656 (63 PTO hp) in 1967 with a hydrostatic transmission, **Fig. 5.50**, and two additional ranges. Other models followed. The central design was developed in cooperation with Sundstrand (USA) [5.4].

The author estimates a total of about 15,000 produced IH hydro tractors. Reasons for terminating this line in 1984 were high initial costs and poor efficiencies.

The design is still exemplary (Table 5.12), as similar configurations are used for the hydrostatic path of modern power split CVTs; however, also with variable bent axis motors which offer lower piston friction forces.

Fig. 5.51 demonstrates typical characteristics of a hydrostatic CVT.

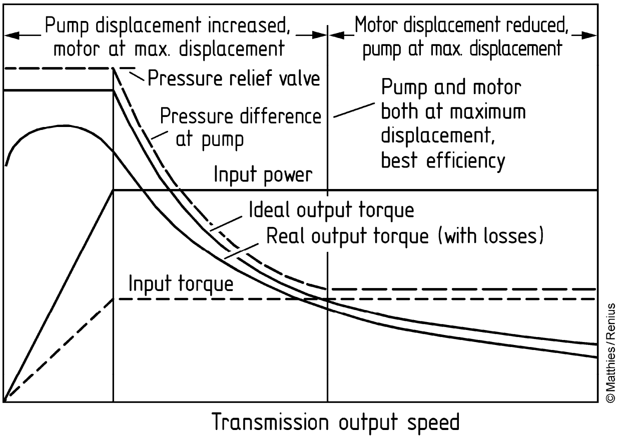

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Fig. 5.51: Characteristics and typical control sequence of displacement for hydrostatic CVTs with variable pump and motor

The author calculated the full load efficiency for the design of Fig. 5.50 based on data from Sundstrand and fundamentals of axle efficiencies, **Fig. 5.52** [5.50].

The efficiency of the hydrostatic unit peaks at 80.5%, still typical for structures with back-to-back swash plate units.

The best full load efficiency of the complete transaxle is, however, only 73% in the low range and 75% in the high range.

This means that it was about 10 percentage points below stepped transmissions and hence too poor for general acceptance.

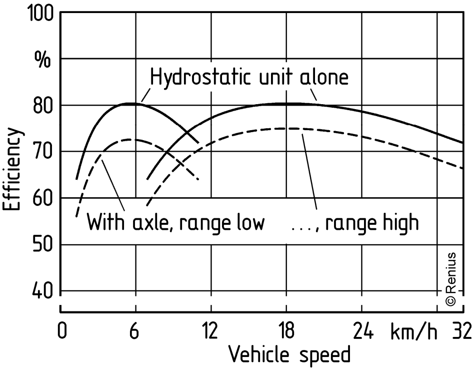

Fig. 5.52: Full load efficiencies for the CVT of Fig. 5.50 and for the complete transaxle [5.50]

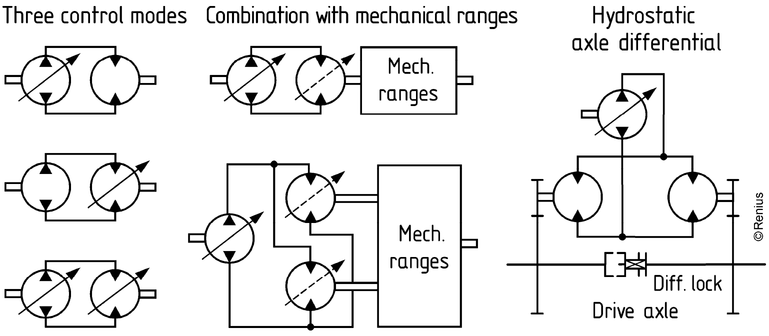

Fig. 5.53: Configurations of direct hydrostatic transmissions

There are several pump and motor configurations possible as shown in **Fig. 5.53**. “Multi-motor” solutions have been applied successfully for loader transmissions [5.61].

One important property of axial piston units for transmissions is their performance when starting from zero rotational speed under load. Swash plate motors have higher friction losses than bent axis units, **Fig. 5.54**. The reason is the very high friction between piston and cylinder [5.62]. The local pressures at the cylinder bushing are so high that there is no separating oil film during starting. This is important for motors in power split transmissions, which achieve their best total efficiencies when the motor is running at very low rotational speeds.

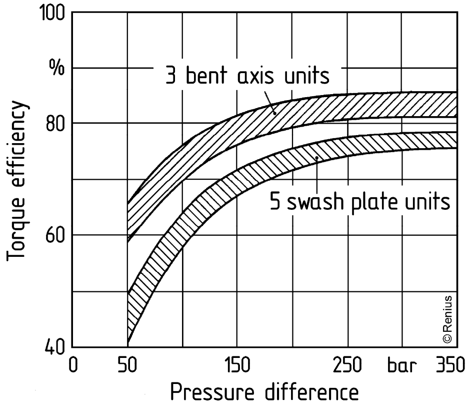

Fig. 5.54: Torque efficiencies of bent axis and swash plate axial piston motors when starting from zero speed under load [5.65]

5.6.4 Electric CVTs

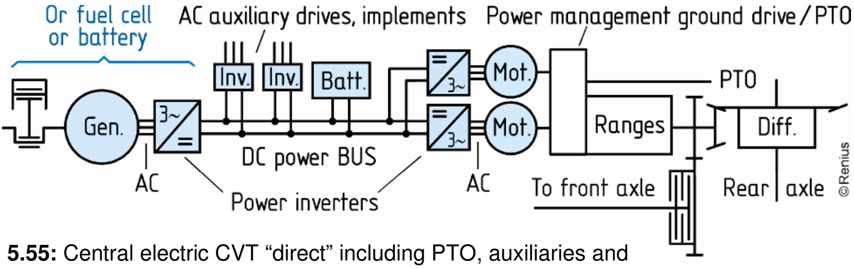

Fig. 5.55: Central electric CVT "direct" including PTO, auxiliaries and implement power supply. Similar early proposals by *M Saller* (MELA)

Three basic concepts are discussed:

1. Central electric CVT, "direct"
2. Individual electric wheel drives
3. Central electric CVT with power split

Fig. 5.55 shows approach (1) with the potential for fuel cells or large batteries or both. Two motors are proposed for variable vehicle speed and variable PTO, addressed in several patents, for example in [5.66].

Table 5.14: Asynchronous Motors ASM and Permanent Magnetic Synchronous Motors PMSM

	ASM	PMSM
Efficiency level	++	+++
Ratio spread	+++	++
Cost per kW	+++	++
Power density	++	+++
Starting from zero	++	+++
Overload capacity	+++	++
Field control	+++	++

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++ moderate +++ good

Design (2), **Fig. 5.56**, offers high flexibility. The author thinks that it needs at least two mechanical power-shifted ranges.

Power split (3) offers the best efficiency level (John Deere patent US 9,562,592 B2).

The DC power bus (e. g. 700V) enables via inverters for all the three structures auxiliary or implement power at variable speeds.

Two common electric units are evaluated in **Table 5.14**. Efficiencies of large PMSM machines are very promising, **Fig. 5.57**. ASM motors are cheaper and improved recently, supported by EU efficiency classes, mainly *class IE4* of EG 640/2009, based on IEC64034-30. ISO 16230-1 defines safety.

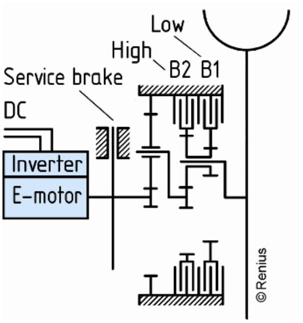


Fig. 5.56: Electric wheel drive with two power-shifted ranges. Proposal of the author

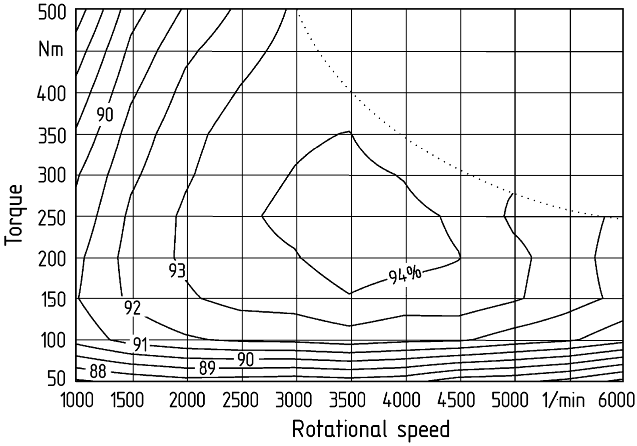


Fig. 5.57: Efficiencies of a very good electric 140 kW PMSM unit with inverter. Courtesy STW

5.6.5 Power split CVTs

Power split can improve total efficiency (at reduced speed spread) or enlarge speed spread (at reduced efficiency) [5.58].

Improved efficiency is achieved by splitting the power into a fixed mechanical path with excellent efficiency and a variable path with usually moderate efficiency. **Table 5.15** classifies power split CVTs by external and internal structures and compact or distributed components. An early internal power split transmission "OC compact" was presented by *Louis Renault* 1907 [5.1]; **Fig. 5.58** shows a similar design [5.61].

Power is divided at the left cylinder block (1) into mechanical power of the rotating housing (3) and hydrostatic power directed to the right cylinder block (2), which is fixed, but its pistons contribute power to the rotating housing. The power fractions are

$$P_{hy} = T_1 \cdot (\omega_1 - \omega_2) \tag{5.12}$$

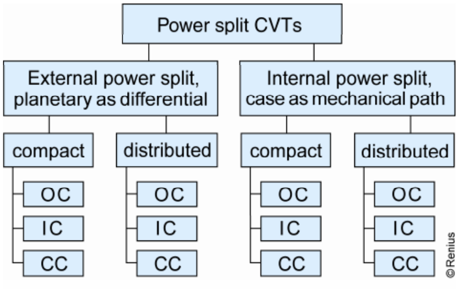
$$P_{mech} = T_1 \cdot \omega_2 \tag{5.13}$$

where

P_{hy} = hydrostatic power
P_{mech} = mechanical power
ω_1 = angular input speed
ω_2 = angular output speed

Design examples of OC, IC, and CC internal power split transmissions "compact" were commented on in [5.67] while a structure OC "distributed" was proposed, **Fig. 5.59**.

Table 5.15: Power split structures: Classification. Definition of OC, IC, and CC see Fig. 5.60


Power split CVTs
External power split, planetary as differential
Internal power split, case as mechanical path
compact
distributed
compact
distributed
OC
IC
CC
OC
IC
CC
OC
IC
CC
OC
IC
CC
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OC output coupled IC input coupled CC compound

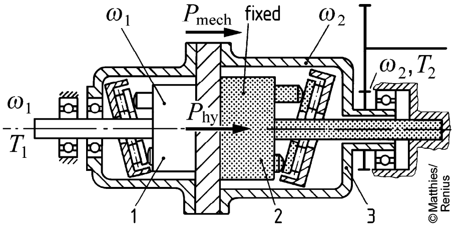


Fig. 5.58: Internal power split, "compact" [5.61]

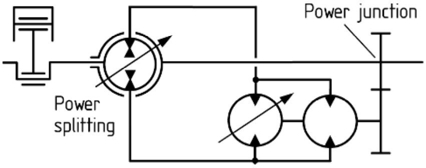


Fig. 5.59: Internal power split, output coupled with "distributed" components [5.65]

Power split transmissions have been classified well by *J.H. Kress*, **Fig. 5.60** [5.68].

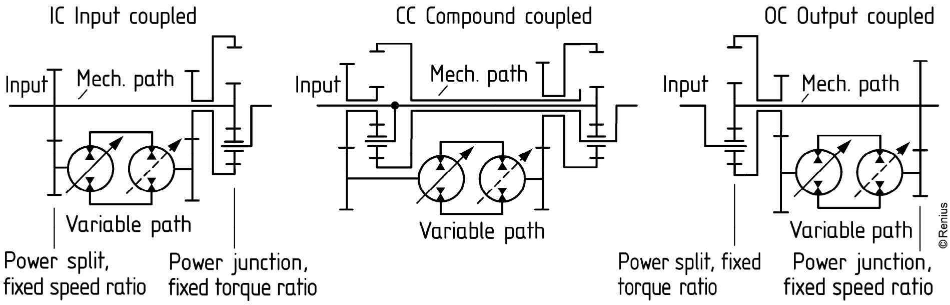


Fig. 5.60: Three basic categories of power split structures, as classified by *James H. Kress*, who also presented complete models in [5.68]. The variable path is at present realized by hydrostatics

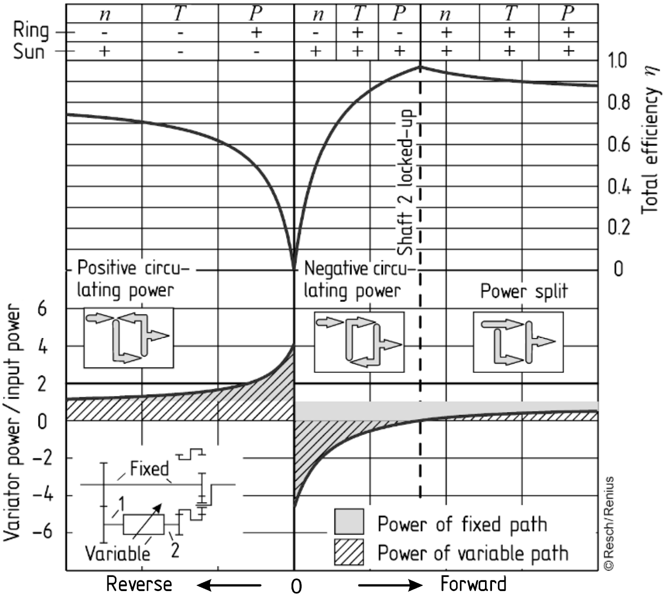


Fig. 5.61: Power characteristics and efficiencies for an *input* coupled IC external power split structure [5.68, 5.50]

The most used structures for tractor drive CVTs (and meanwhile also for wheel loaders and other mobile machines) are those as addressed in Fig. 5.60.

Characteristics of IC and OC structures are analyzed in **Fig. 5.61** and **Fig. 5.62**, based on [5.68] and [5.50].

The following efficiencies are assumed for both diagrams:

- CVT unit: $\eta = 85\%$ (45° bent axis units)
- gears, bearings, seals: $\eta = 97\%$ (in total)

These values address full load, which is, however, not possible in reality for speeds below the traction limit.

But in order to simplify the analysis, calculation has been done with the listed efficiencies also for the low speeds with nominal power.

Regarding the table at the top of the diagrams, the following sign convention is used:

- Speed n positive, if direction of input speed
- Input P power positive, output power P negative
- Sign of torque T resulting from speed and power.

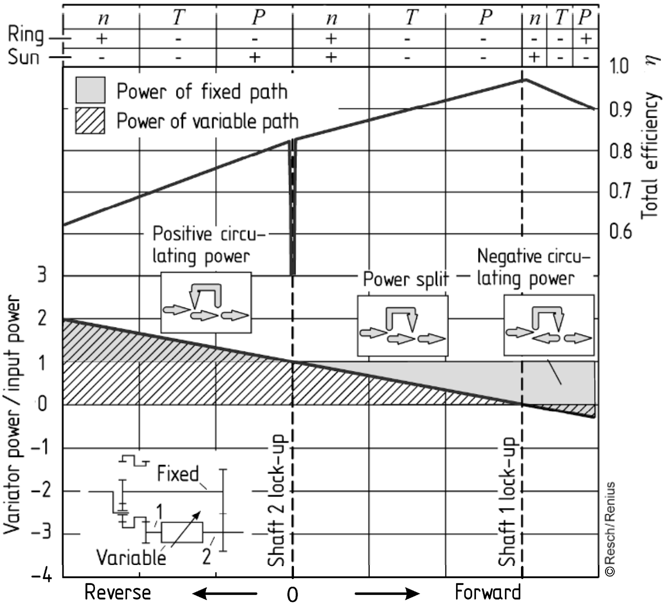


Fig. 5.62: Power characteristics and efficiencies for an *output* coupled OC external power split structure [5.68, 5.50]. Efficiency of variable unit constant 85% and of all mechanical elements together constant 97% for both diagrams

Total CVT efficiency depends mainly on the efficiency of the hydrostatic unit and the output-to-input ratio.

Fig. 5.63 demonstrates the relationships for an input coupled structure as in Fig. 5.61.

Acceptable total efficiencies are only possible near the lock-up point. This is the reason that power split CVTs of this kind need additional mechanical ranges. The influence of the variator unit is not so high within the recommended ratio, which allows compact and low-cost units.

The total efficiency of the OC structure is completely different, **Fig. 5.64**. If efficiency of the variable unit is at least 0.85 (possible with variable 45° bent axis units for pump and motor), the total efficiency forward is very flat and additional mechanical ranges are not a must. The use of reverse speeds is possible, but high reverse speeds create high losses owing to circulating power, Fig. 5.62.

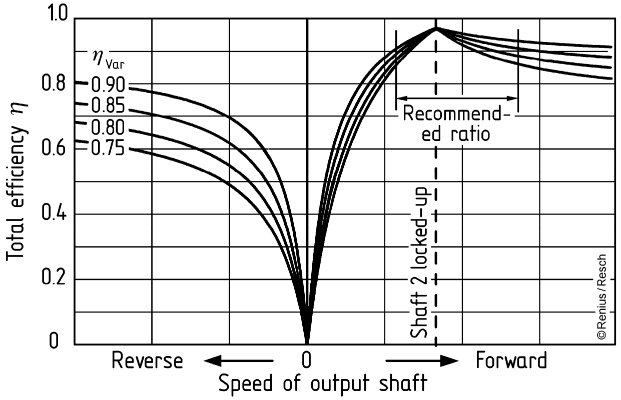


Fig. 5.63: Influence of the variator unit efficiency on the overall efficiency for the IC structure of Fig. 5.61 [5.50]

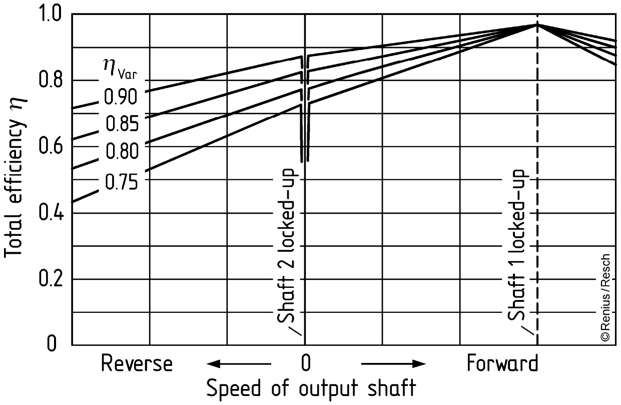


Fig. 5.64: Influence of the variator unit efficiency on the overall efficiency for the OC structure of Fig. 5.62 [5.50]

The third group of compound coupled power split CVTs (CC, see Fig. 5.60) is much more complicated and thus not analyzed in detail in this book.

However, a recommended survey has been published in [5.69] organizing the huge variety into 3 basic groups, **Fig. 5.65**.

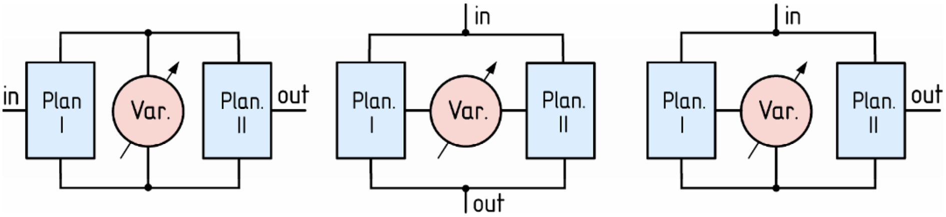


Fig. 5.65: Basic structures of power split CVTs with compound planetaries, after *Gackstetter* [5.69]

5.6.6 Examples of power split CVTs

A very first important approach was the famous internal power split design proposal of *Louis Renault* in 1907 for passenger cars as cited in chapter 5.1.

Beginnings of a new era. The discussion on tractor power split CVTs was not in the foreground for a long time but in 1985 addressed by a patent of the Fendt engineer *Hans Marschall* (1936–1989) [5.70] and a patent of the German professor *Friedrich Jarchow* (1926–2011) [5.71] as licensed by CLAAS. Both companies confidentially developed their own power split CVTs based on these patents. At that time, the CVT efficiency was still an important question:

Will it be able to compete with efficiencies of full power shift transmissions?

The author was requested by CLAAS and Fendt around 1990 to define a target facing this question, **Fig. 5.66**.

Some years later, the target was released for publication, first in 1993 [5.72]. Taking in account full load efficiencies of the axles of about 94–95% (Fig. 3.30 and 3.31) and a full load efficiency of 98.5% for one additional gear wheel engagement, the target of Fig. 5.66 requires *full load "gearbox" efficiencies of 88–90%. This was commented on by experts as being extremely challenging.*

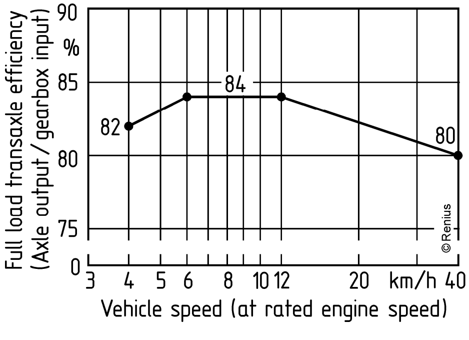


Fig. 5.66: Full load efficiency target (axle output/ gearbox input) for tractor CVTs of nominal engine power above about 100 kW [5.72]

However, several power split CVTs of large tractors (above 100 kW) could meet this specification; the first one was the CLAAS HM8 [5.73].

Pioneering work has been carried out since the late 1980s by CLAAS Industrietechnik, resulting in the CVTs CLAAS HM-8 (about 140 kW) [5.36] and the later larger CLAAS HM-II, also called "Traxion" (about 220 kW). Both were developed for the CLAAS carrier tractor "Xerion", both based on the patent of *F. Jarchow*.

The HM-8 had seven power split ranges in order to keep the hydrostatic power portion very low in favour of high efficiency. One additional creeper range worked directly. This CVT has been produced since 1996 in small numbers. The follower HM-II (1999) had only five power split ranges and a conventional power reverser. It offered improved shift comfort but could also only attain at a small production volume.

The CLAAS-Jarchow approach was characterized by an input coupled four-shaft compound planetary consisting of two standard planetaries [5.36]. The shifts between the ranges were done at synchronized speeds with *dog clutches* and the power handed over *without interruption* in a very short time, during which the ratio was, however, kept constant (no acceleration or deceleration of the vehicle). The power shift by dog clutches was one reason for good efficiencies but it was finally not possible to arrive at a 100% shift reliability with this principle.

So CLAAS decided to use a ZF CVT for the Xerion line. However, the collected experience was later on very helpful for the CLAAS EQ200, produced from 2014.

The idea of a dog clutch range power shift was realized with more success by the Steyr "S-Matic", attaining series production for Steyr and Case IH tractors in 2000 [5.50, 5.74]. The CVT producer "Steyr Antriebstechnik" became part of ZF in 2000.

As used for the **Fendt "Vario" CVT**, the structure of the CVT in *Hans Marschal's* patent [5.70] is an advancement over proposals of *Hans Molly* (1902–1994) [5.75].

The "Vario", **Fig. 5.67**, was presented in 1995 at AGRITECHNICA and first applied to the Fendt "Favorit 926", the largest Fendt tractor at that time. It was the first power split tractor transmission in the world to achieve series production, starting in Marktoberdorf in 1996 [5.76]. The output coupled structure requires outstanding efficiencies of the hydrostatic units (Fig. 5.64).

Swash plate units were not appropriate and also had the disadvantages of limited tilting angles (max. 20–22°) and very high piston friction at low speeds. Bent axis units had better efficiencies (mainly for low speeds, Fig. 5.54) and higher swivel angles, but were also not good enough for the "Vario". Even all commercially available variable axial piston pumps and motors had been quoted as too poor in efficiency to meet the required level. But there was hope for improving efficiency for bent axis units, based on measurements at TH Braunschweig by *D. Hoffmann* for *fixed* Flygmotor units in 1970 [5.77]. He explained outstanding efficiencies by the large 40° angle and the spherical pistons with piston rings; however, *variable* "wide angle" versions did not appear on the market.

Fendt did something unusual, developing on their own a new variable bent axis unit with a 45-degree maximum swivel angle, **Fig. 5.68** [5.76]. A co-operation with Sauer-Sundstrand (2013 Danfoss) supported series production. This strategy was able to meet the target, **Fig. 5.69**. Full power shift transmissions can be slightly better but the "Vario" can compensate small differences by its infinitely variable electronic ratio control being superior to a stepped ratio control, as confirmed by many DLG tests. Since 2009, Fendt has produced all tractor models with the "Vario" only (Status 2019).

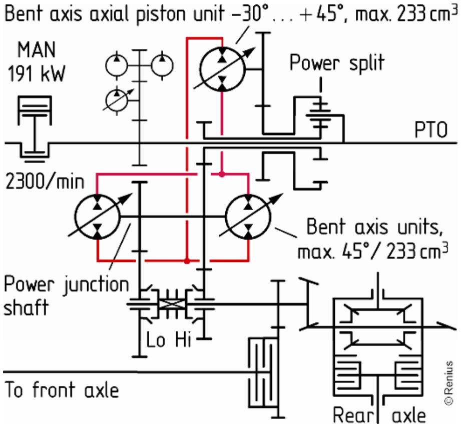


Fig. 5.67: Structure of the Fendt "Vario", the first mass-produced tractor power split CVT, presented at AGRITECHNICA 1995, in series as of 1996

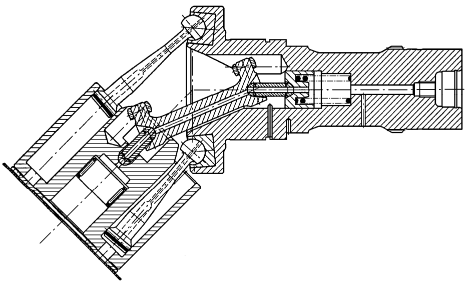


Fig. 5.68: Variable bent axis axial piston unit with 45° max. swivel angle [5.76]

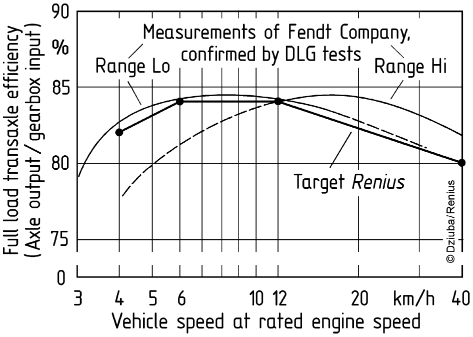


Fig. 5.69: Full load efficiency of the Fendt "Vario", size ML 200 for the "Favorit 926" (engine power 191 kW), and comparison with a target [5.76]. High speeds profit from the low hydrostatic power

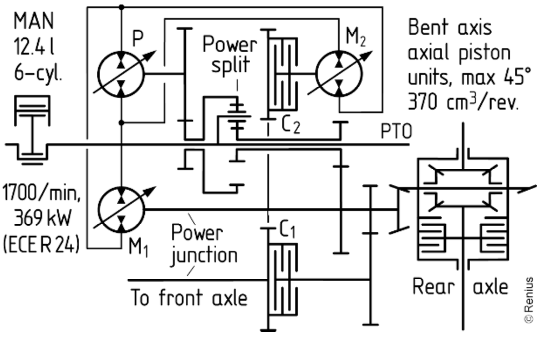
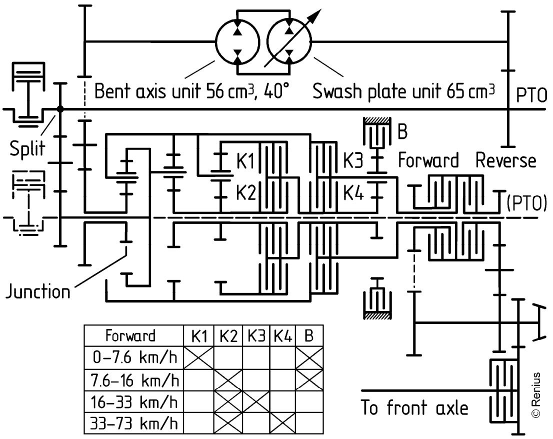


Fig. 5.70: New Fendt Vario. Series production from 2015 with the new 1000 tractor family, max. 380 kW [5.77], in 2019 also introduced to the new 900 line



Forward	K1	K2	K3	K4	B
0–7.6 km/h	☒				☒
7.6–16 km/h		☒			☒
16–33 km/h		☒	☒		
33–73 km/h		☒		☒	

Fig. 5.71: Second generation of the ZF "Eccom", now "Terramatic", size 28–32 [5.20], 2013 first for CLAAS, up to about 200 kW, size 42 later up to 320 kW (2019)

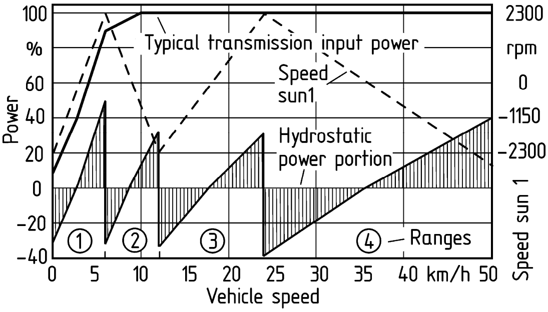


Fig. 5.72: Transmission input power, hydrostatic power portion and speed of the first sun gear of the first ZF CVT "Eccom" [5.79]. Terramatic (Fig. 5.71) is similar

The second generation of the Fendt Vario CVT is an advancement over the first design, also output coupled, **Fig. 5.70**. The pump P drives the hydro motors M1 and M2. When the clutch C1 is open, the parallel working hydro motors produce a longitudinal differential effect, for which motor M2 drives via C2 only the front axle. C1 can be closed to function as a differential lock.

This structure is now able to offer a pull-in-turn function and a permanently engaged four-wheel drive on the road without any wind-up effects [5.78].

The first power split CVT of ZF was the "Eccom", introduced with Deutz tractors in 2001 [5.79]. A second similar generation came on the market in 2014, first in CLAAS tractors, **Fig. 5.71**. All ZF CVTs are input coupled with a compact hydrostatic unit of a variable pump and a fixed motor in a back-to-back design, in the first generation with two swash plate units, now also with bent axis motors. The hydrostatic power portion is moderate, **Fig. 5.72**.

As swash plate units have high piston friction near zero speeds (see Fig. 5.54), the motor was replaced in the larger Terramatic versions by a bent axis unit. This improves both, control quality and efficiency. However, influence on the losses is not high because of low hydro power.

The advantage of low costs for the hydro unit must be paid for by the disadvantage of small ratio spreads. The Terramatic has four ranges for top speeds 40/50/60 km/h. It is offered in many versions. The ranges are power-shifted in both CVTs of Fig. 5.71 and 5.72 under synchronized conditions.

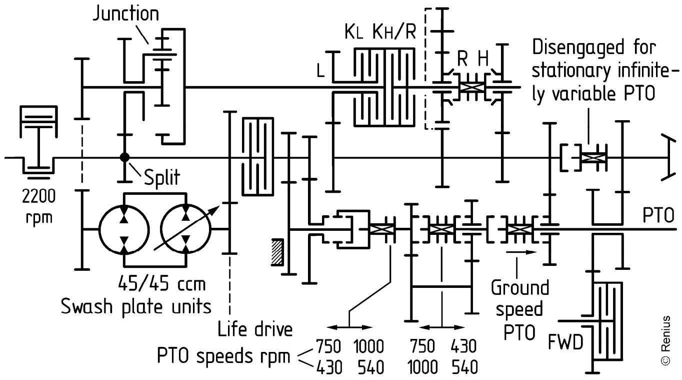


Fig. 5.73: The smallest CVT of the ZF Terramatic family is the type TMT 09, first introduced for the tractor “Lintrac” (2014) of the Austrian tractor manufacturer Lindner. Nominal engine power 74 kW. Complex PTO structure

For the lower end of its CVTs, ZF developed the “Terramatic” sizes 09 and 11 (max. 85 kW). Type 09 was introduced in 2014 with the “Lintrac 90” (40 km/h), **Fig. 5.73**. It has two ranges forward and one reverse, power-shifted by a dual clutch. The transmission offers four PTO speeds, a ground speed PTO, and a stationary CVT-PTO. The main losses are generated by the hydrostatic path, **Fig. 5.74**, in particular at left due to circulating power (Fig. 5.61 and 5.72).

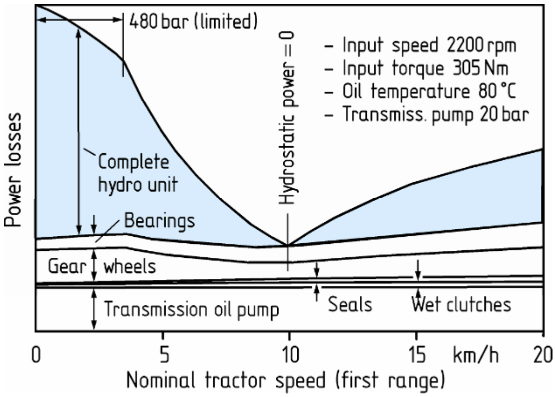


Fig. 5.74: Calculated power losses for the first range of the ZF Terramatic 11 (identical with Fig. 5.73) [5.80]. Lowest losses at 10 km/h as hydrostatic power is zero

JD presented its first own CVT in 2001, **Fig. 5.75**. It works with two bent-axis axial piston units from Danfoss, which are similar to those of the Fendt “Vario”; however, in a completely different structure. It is input-coupled and uses a four-shaft compound planetary. The engine drives both, the variable axial piston unit and the far left sun gear.

The speed of the left ring gear is controlled by the speed of the fixed displacement hydro unit. Its carrier output is used for range L (KL) and output of the mid sun gear for H (KH, max. 50 km/h).

Forward range shifting is done at synchronized speeds. An extra planetary for reverse uses the carrier output of the merging planetary; reversing is achieved by engaging the brake KR, output at its sun gear.

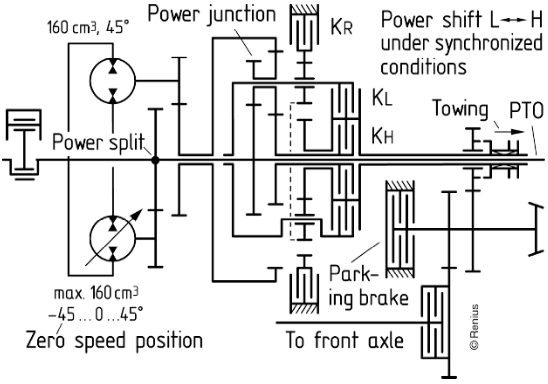
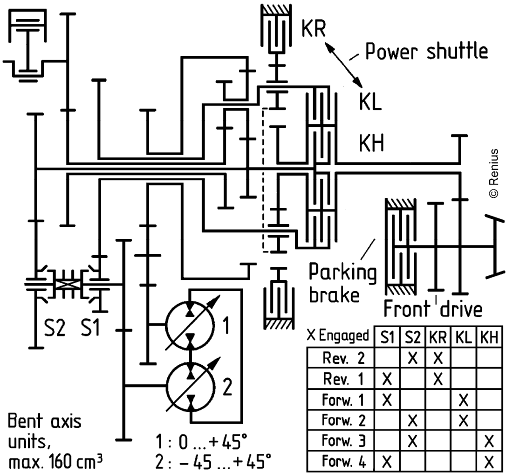


Fig. 5.75: First power split CVT designed by John Deere, presented in 2001 for the tractors 7710 and 7810 (118 und 129 kW), later also for the 7020 and 7030 models



X Engaged	S1	S2	KR	KL	KH
Rev. 2		X	X		
Rev. 1	X		X		
Forw. 1	X			X	
Forw. 2		X		X	
Forw. 3		X			X
Forw. 4	X				X

Fig. 5.76: John Deere AutoPowr for the upper 8030 tractor family (max. 236 kW), presented in 2006

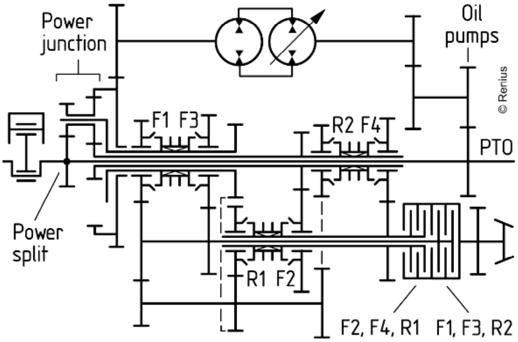


Fig. 5.77: CNH CVT with automatic dual clutch range selection (2009), nominal power up to 185 kW

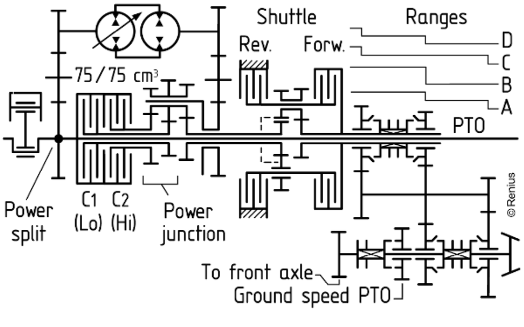


Fig. 5.78: Power split CVT of Valtra (AGCO), as presented in 2009 within a modular transmission system [5.81]

In 2006, John Deere presented the power split structure of **Fig. 5.76** for its large 8030 family (max. 236 kW).

In its right part, it is similar to the AutoPowr of the 7000 series with an identical power shuttle (Fig. 5.75); in its left part, however, output coupled. The engine drives the left sun wheel of the planetary gear set; power is split at the dual planetary gear. The ring gear powers the hydrostatic unit, which drives the planetary carrier (via S1) or the mid sun gear (via S2).

In 2009, CNH introduced a CVT with an interesting range shift, **Fig. 5.77**. It is input-coupled and has a compact hydro unit in a back-to-back configuration but in this case applies the dual clutch principle, which allows realizing four power shifted ranges forward (F) and two reverse (R) with only two clutches.

The same principle was realized by CNH 2013 in a lower power class; however, in this case with two ranges only, see Fig. 5.83 below.

At SIMA 2009, Valtra (AGCO) presented an interesting modular transmission system for tractors in the upper power range (series N and T) which realizes both, a continuously variable power split CVT and, as an alternative, a stepped version with five power-shifted speeds and with many identical parts; see both transmission maps as published in [5.81].

The CVT version is input-coupled, **Fig. 5.78**. The power junction takes place at the threefold planetary unit: Input with engine speed either at the mid or the left sun (C1, C2) and input of the hydrostatic path at the common carrier. Output is at the right sun wheel, followed by the power reverser (rotating forward as a block) and four synchronized ranges.

SAME Deutz-Fahr uses ZF CVTs but has also developed its own power split CVT for the lower power range including narrow track tractors, **Fig. 5.79**. It has no ring gears or outside control brakes and thus represents a very slim design achieving a narrow shape.

The transmission is input-coupled; the range L uses the left planetary carrier (KL) and the range H its right sun gear (KH). Reverse is realized by opening KL (or KH) and closing KR. The right planetary set is in this case connected with the left carrier (countershaft case).

Power split versions are becoming popular in **Japan** as well, even for the low engine power levels, **Fig. 5.80**. This Kubota CVT is input-coupled, has a power reverser and three additional ranges. The PTO offers five speeds and one reverse. The front axle drive uses a pull-in-turn overdrive (chapter 3).

In 2011, John Deere presented another own power split transmission, **Fig. 5.81**. It has a compound planetary and works with three automatic ranges and a synchronized reverser. The double yoke bent axis unit from Danfoss with 45° tilting heads saves costs. Efficiency losses by the mechanically connected tilting heads may be compensated for by short pressure pipes (low losses).

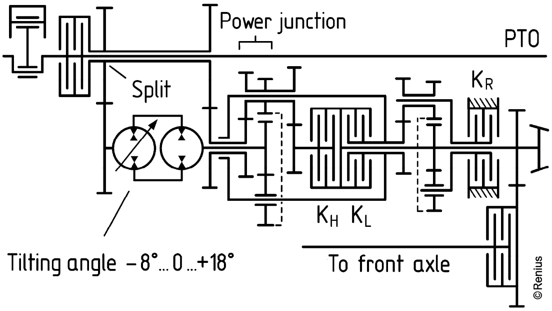


Fig. 5.79: Same-Deutz-Fahr power split CVT for lower power ranges including narrow track tractors (2010)

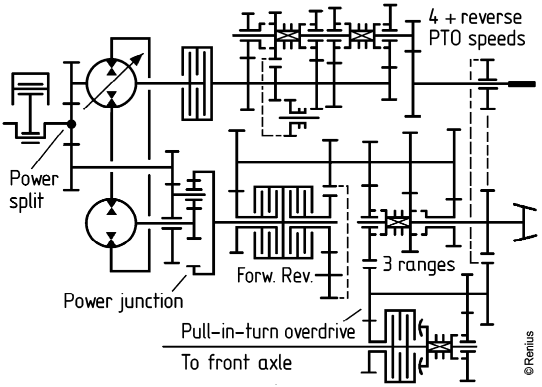


Fig. 5.80: Kubota power split CVT for small tractors, power range of 23–43 kW, introduced 2011

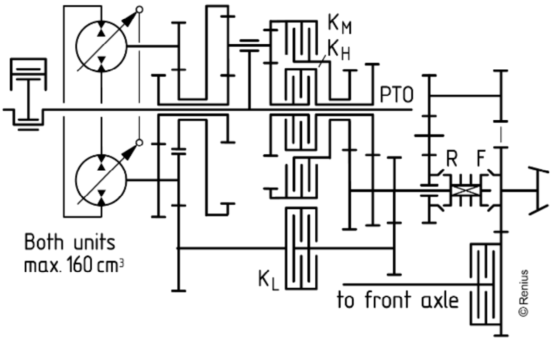


Fig. 5.81: John Deere AutoPowr with the Danfoss 45° double yoke bent axis unit (combined tilting heads) for the tractor family 7R (147–206 kW ISO, 2011)

The common carrier for the left and the right planetary rotates with engine speed. Its left part is connected to the hydrostatic unit via the ring gear and the sun gear. The speed relation of both is controlled by the hydro unit and generates the rotational speed of the double planetary gear wheel. Its speed is combined with the carrier speed, generating the speeds of the right ring gear and sun gear and thus the vehicle speed for M and H. Only the low range (KL) is output-coupled.

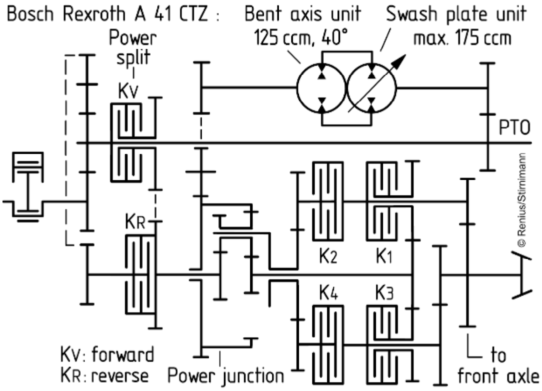


Fig. 5.82: Power split CVT of CNH Industrial for high power tractors NH T8000 and Case IH Magnum (2013)

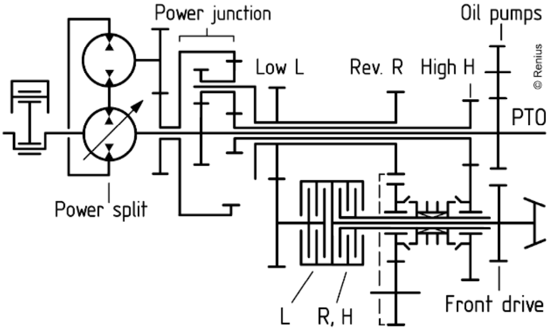


Fig. 5.83: The small dual clutch power split CVT of CNH, only three ranges. Introduction 2013 for Case IH Maxxum, New Holland T6, and lower NH models T7

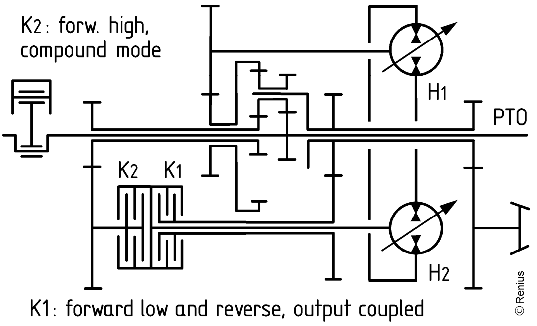

K2: forw. high, compound mode
H1
PTO
K2
K1
H2
K1: forward low and reverse, output coupled
© Renius

Fig. 5.84: CLAAS power split CVT EQ 200 for Arion 500 and 600 tractors (2014). From 2017, an additional version EQ 220 has a second double clutch package for a direct hydrostatic reverse mode

For their high-power tractors NH T 8000 and Case IH Magnum, CNH Industrial developed a power split CVT, which does not use the double clutch principle (as applied to two other CVTs from CNH), **Fig. 5.82**.

The structure is input-coupled with a reverser between engine and split point. The planetary gearing delivers two output speeds, "low" at the carrier and "high" at the right sun. The clutches K1–K4 belong to the four ranges. The fixed 40° *bent axis* motor offers slightly improved efficiency over that of an axial piston *swash plate* motor.

In 2013, CNH introduced the dual clutch power split CVT of **Fig. 5.83**, It is similar to that of Fig. 5.77; however, with half as many ranges. The two forward ones are automatically shifted at synchronized points, the reverser L–R is power-shifted.

In 2014, CLAAS introduced its power split CVT EQ 200, **Fig. 5.84**. It has some roots in the former CVT project SHL of Voith with further genes from *P. Dziuba* supporting a CVT development under *Marcus Liebherr* (1948–2010). The final design of CLAAS with 45° Danfoss bent axis units incorporates a modern electronic CLAAS control system.

The first range (forward-reverse) works output-coupled like the Fendt Vario: K1 is closed and the hydrostatic unit H1 is driven by the ring gear, while the shift to range 2 (K2) reverses the hydrostatic power flow.

Electric power split CVTs offer small further benefits in efficiency, but their DC power bus (Fig. 5.55) can drive implements etc.

Such developments were announced late 2019 by John Deere (patent US 9,562,592 B2) and by ZF.

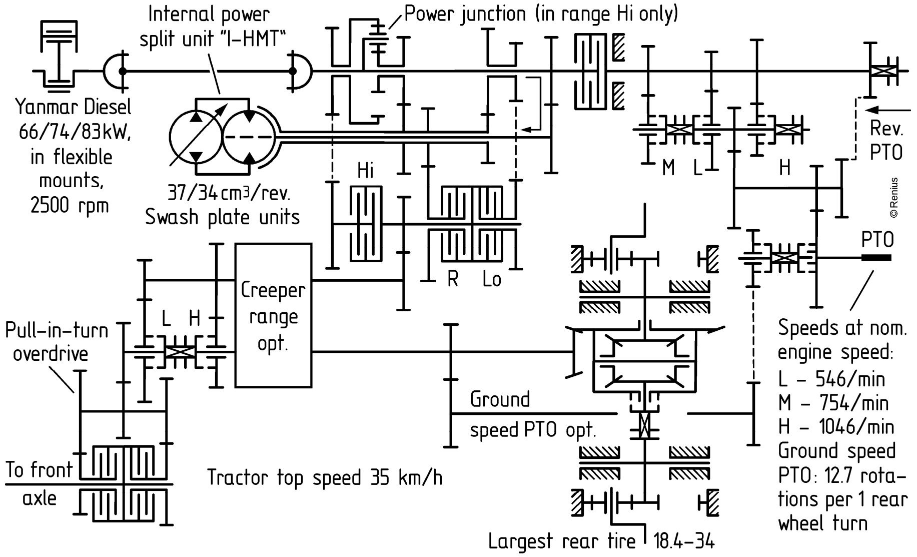

Internal power split unit "I-HMT"
Power junction (in range Hi only)
Yanmar Diesel 66/74/83kW, in flexible mounts, 2500 rpm
37/34cm3/rev. Swash plate units
Hi
R
Lo
M
L
H
Rev. PTO
©Renius
PTO
Creeper range opt.
L
H
Pull-in-turn overdrive
To front axle
Ground speed PTO opt.
Tractor top speed 35 km/h
Largest rear tire 18.4-34
Speeds at nom. engine speed:
L - 546/min
M - 754/min
H - 1046/min
Ground speed PTO: 12.7 rotations per 1 rear wheel turn

Fig. 5.85: Power split transmission of the Yanmar tractors YT490, YT5101, and YT5113 (Japan 2015) using the internal power split unit "I-HMT", principle first for tractors by Yanmar in 2009

An **internal power split CVT** was proposed as early as 1907 by Louis Renault [5.1]; however, with no breakthrough.

A first use in series for tractors took place in 2009 by Yanmar (Japan) for their tractor ranges EG 300 and EG 400 (26–53 hp), based on US patent 8,302,525 B2 (filed Nov. 21, 2006). The company reports considerably higher efficiencies over "direct drive" hydrostatic transmissions (HMT), encouraging larger versions, **Fig. 5.85**.

The diesel engine drives the CVT via the inner shaft (arrow). The outer output shaft is connected with the double clutch delivering the ranges R and Lo. The Hi range works with a second (external) power split: The output of the I-HMT is led to the sun gear of the upper planetary while the carrier contributes engine power, both combined at the ring gear.

The three power-shifted basic ranges L, H, and R are combined downstream with two collar-shifted ranges L and H. A creeper range is available as an option. The 4WD works with a pull-in-turn overdrive.

The transmission structure enables five PTO modes including a reverse and a ground drive PTO. The internal power split unit is input coupled; displacements are realized by the swash plate principle, **Fig. 5.86.** The input shaft drives the cylinder block. The left stationary swash plate controls the ratio; the right-hand one delivers the output speed at the reaction torque of the piston slippers. The connection between the left and the right cylinder chambers is coordinated by a special internal timing valve.

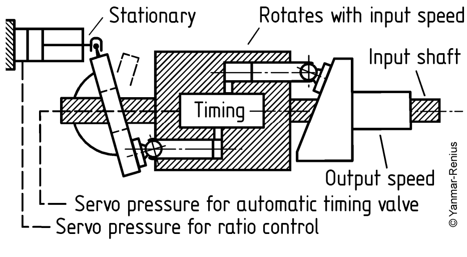


Fig. 5.86: Principle of the Yanmar internal power split unit "I-HMT", used for the YT tractor series. Courtesy Yanmar (Japan)

5.7 Transmission Components

5.7.1 Clutches and shift elements

The cheapest concept for a **master clutch** and **independent PTO clutch** is a combination of both, **Fig. 5.87**. A diaphragm spring (with declining force increase versus travel) is under preload giving the necessary clamping force. Torque can be calculated with the effective radius r_m (r_m from equ. 3.6) and typical friction coefficients of about 0.35–0.40 for dry linings. **The nominal slip torque** for moderate torque back-ups should be about 250% of rated torque for the ground drive clutch and slightly less for the PTO clutch.

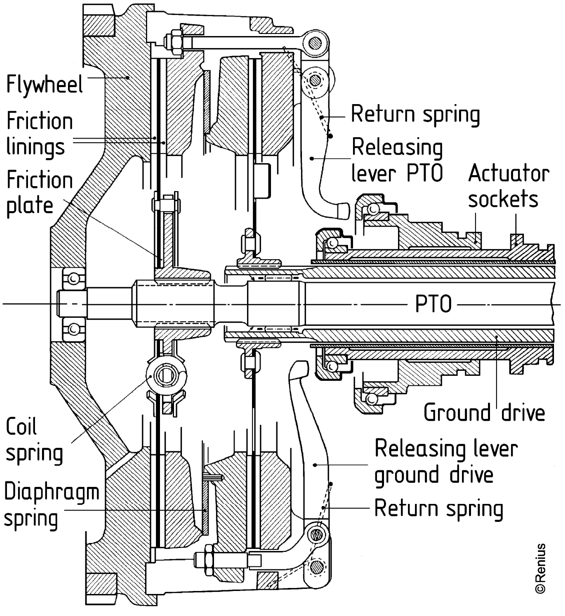


Fig. 5.87: Combined dry clutches for driving and PTO, cheapest solution for independent PTO Courtesy LuK

Hydrodynamic couplings have been used for some decades by some tractor companies in addition to a dry master clutch, mainly by Fendt (Germany), **Fig. 5.88**. They allow a very comfortable drive-off by only increasing the engine speed and they reduce dynamic torque loads on the whole transmission [5.82, 5.83].

The power losses are low in the case of high engine speeds and low loads, **Fig. 5.89**, on average for a multi-purpose tractor 2–4%, the same for heavy full power operations.

Fendt used the stored performance map for modulating power shifting of a transmission (Fig. 5.33) for high shift comfort.

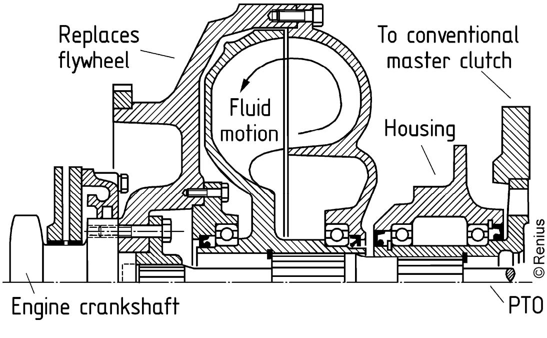


Fig. 5.88: Hydrodynamic coupling between engine and master clutch. Based on material of Fendt

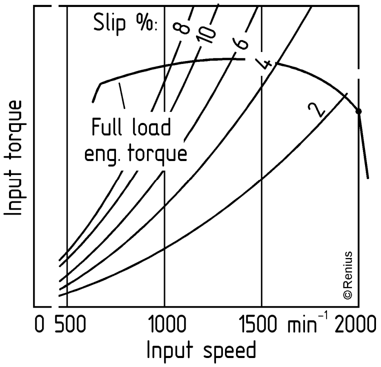
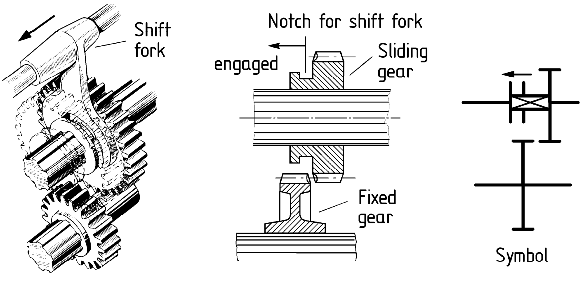


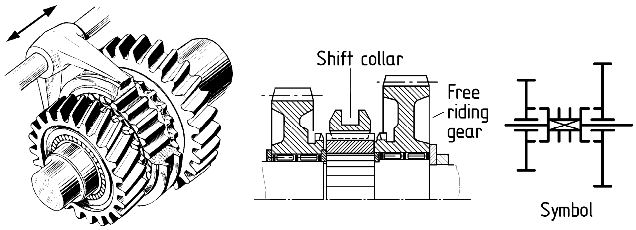
Fig. 5.89: Performance of a hydrodynamic coupling. Courtesy *F. Görner*, Fendt

Table 5.16: Design of the most important structures for manual gear shift with power interruption. 3-D graphics courtesy ZF

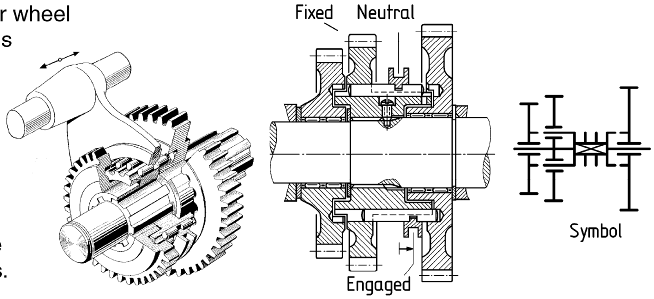
Shift by sliding gear is the cheapest shift design by far, as the function of shifting is added to the upper gear wheel with almost no additional costs. Versions with synchronized shift are not possible.
Typical application: Reverse gear wheels for technology levels I-II.



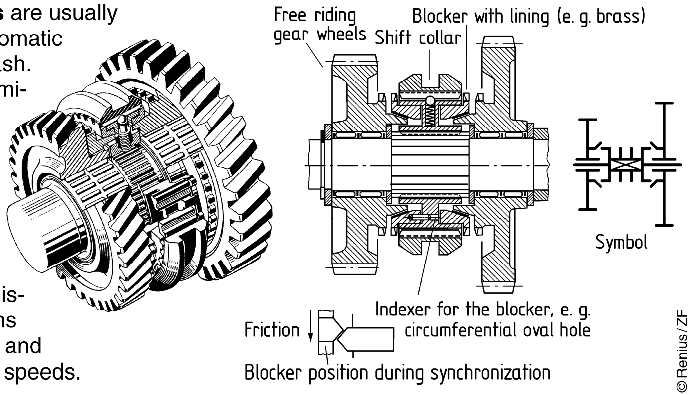
Collar shift with involute profiles is very often used for lower technology levels. Shift comfort is very limited but synchronized shift is possible as an option, more for basic speeds than for ranges.



Pin shift through a gear wheel can save space, which is mainly of interest for reverse speeds, as shown in Fig. 5.12. However, it requires outstanding precision in meeting the very low tolerances of the circular pitch and the diameters of the bores.



Synchronized shifts are usually designed with an automatic blocker to prevent clash. The shift collar has limited rotational play. When it is operated, its friction force rotates it less than one pitch, blocking the shift. Shifting is done automatically when the friction moment disappears. This happens when the gear wheel and the collar have same speeds.


Free riding gear wheels
Blocker with lining (e. g. brass)
Shift collar
Symbol
Indexer for the blocker, e. g. circumferential oval hole
Friction
Blocker position during synchronization

©Renius/ZF

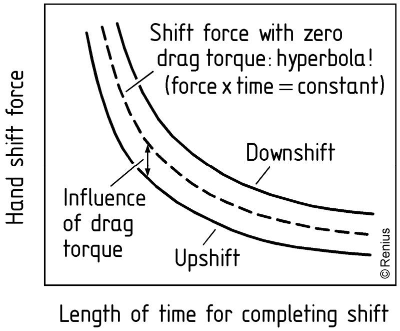


Fig. 5.90: Typical relationship between shift force and shift time of a synchronizer

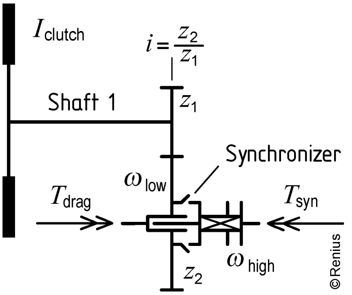


Fig. 5.91: Synchr. downshift: T_{syn} balances the clutch plate acceleration torque *plus* drag torque

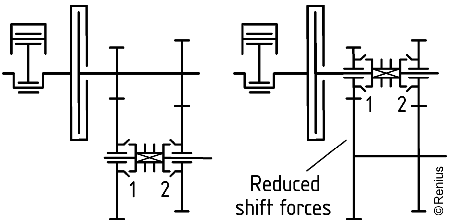


Fig. 5.92: Position of a synchronized shift unit near the master clutch prevents the i^2 effect

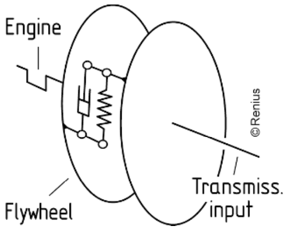


Fig. 5.93: Damper at transmission input

Dimensioning of synchronizers has to consider two basic criteria:

1. *Shift force* (functionality, comfort)
2. *Thermal load* (durability, wear)

Downshift is always more critical than upshift, **Fig. 5.90**, as explained below.

Force x time = Shift quality index

Example downshift, **Fig. 5.91**: the synchronizer must accelerate shaft 1. The mass inertia of the master clutch usually dominates; other mass inertia can be neglected. The clutch inertia must be converted to the lower shaft by i^2. The downstream shaft represents the large tractor mass not being influenced by the shift.

Synchronization time t_{syn} for shaft 1 for constant forces during the shift is

$$t_{\text{syn}} = \frac{\omega_{\text{low}} - \omega_{\text{high}}}{T_{\text{syn}} - T_{\text{drag}}} \cdot I_{\text{clutch}} \cdot i^2 \qquad 5.14$$

where

ω_{low} = angular velocity of low speed side
ω_{high} = angular velocity of high speed side
T_{syn} = friction torque of the synchronizer
T_{drag} = drag torque at the synchronizer
I_{clutch} = mass inertia of master clutch
i = ratio of gear engagement(s) upstream

In the case of an upshift the sign of the drag torque is positive explaining the lower curve in Fig. 5.90 (drag torque "helps").

Rule: The i^2 effect recommends locating the synchronizer at the upper shaft, ***Fig. 5.92****.*

The *thermal load* depends on the type of synchronizer, the shift force, the coefficient of friction, and the cooling (cooling in tractor transmissions often by using forced lubrication systems). Units with multi-layer friction elements can further increase capacity.

Modern diesel engines have a high torque pulsation not fully compensated for by the flywheel. The main reasons are high power density and low nominal speeds. Dampers are common, **Fig. 5.93**. They work with friction or (better) with hydrodynamic damping elements.

Shift mechanisms of manual tractor transmissions are similar to those of passenger cars or trucks but have to operate more speeds. A typical schematic configuration for six basic speeds, **Fig. 5.94**, has a shift lever which operates the three sliding selector shafts via the indexing shaft.

Tractors of the lower technology levels I and II often use central shift levers (dashed) being the cheapest design while the offset position (right hand of the driver position) is typical for slightly more comfort and cabs with a flat bottom, see also the ZF transmission of **Fig. 5.95**.

Critical synchronized shifts of tractor transmissions (mainly for range and reverse shifts with high i^2 terms) need high operating forces (equ. 5.14). High reduction ratios of the linkage reduce hand forces but increase shift lever travels.

Travel ratios of about 6 to 10 are chosen as a compromise, upper values if the synchronizer unit offers short shift travels [5.84]. All contacts of the linkage should be surface-hardened.

It has been noted that a **pin shift through a gear wheel** (Table 5.16) can save space and costs, but can be critical in production regarding the necessary tolerances. In the case that they are not perfectly machined, the pins are not loaded equally, and they can be deflected under peak loads with the danger of popping out after some time. However, the possibility of shifting through a gear wheel can be attractive.

An interesting alternative design is used by ZF, **Fig. 5.96** (element of Fig. 5.30). Low circumferential pitch tolerances can be machined easily with involute spline profiles, a standard design for shift collars.

The right hand shift collar has two sections of involute splines which are moved axially through the two "windows" of the left gear wheel engaging with an idler gear (not visible in Fig. 5.96).

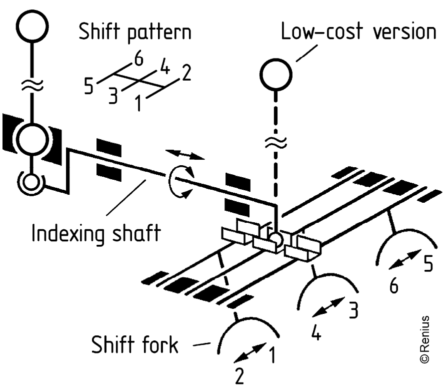


Fig. 5.94: Shift mechanism for six speeds with two possible lever positions

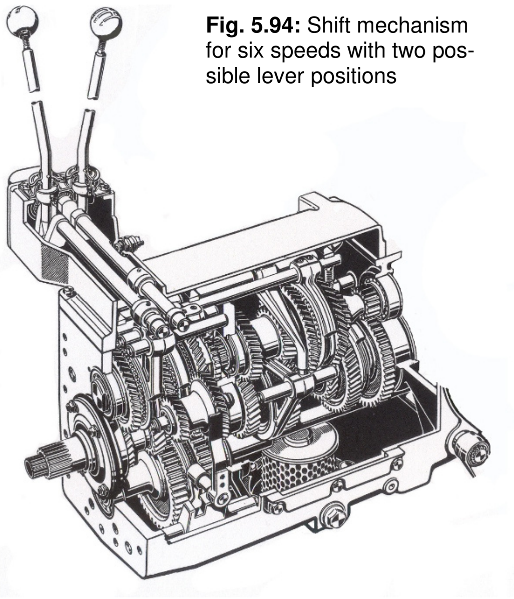


Fig. 5.95: Shift mechanism for six speeds forward (right) and two split ranges and reverse (left). Courtesy ZF (transmission T 3000, 1970)

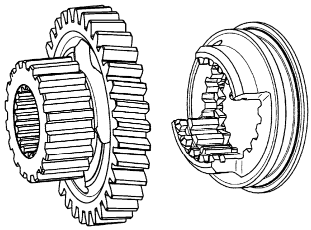


Fig. 5.96: Collar shifting through gear wheel "windows" by two stems. Courtesy ZF

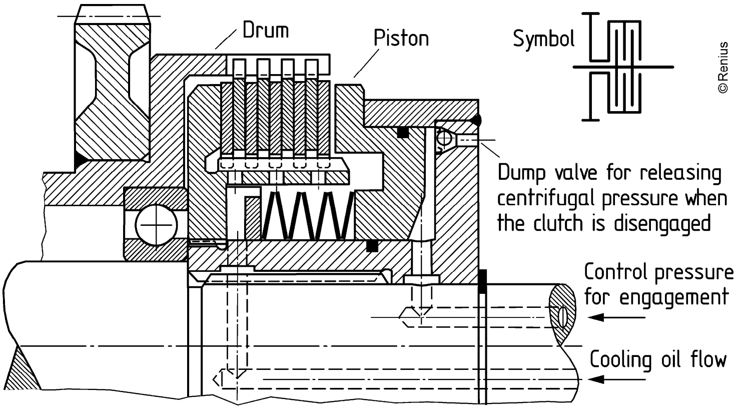

Drum
Piston
Symbol
©Renius
Dump valve for releasing centrifugal pressure when the clutch is disengaged
Control pressure for engagement
Cooling oil flow

Fig. 5.97: Wet multiple plate clutch, important for power shifted tractor transmissions. The shifting is done by hydraulic pressure (often about 20 bar = 2 MPa = 290 psi), disengagement by coil springs

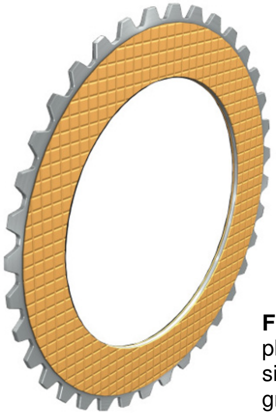


Fig. 5.98: Typical plate with layer of sinter bronze and grooves for cooling and fluid draining.

Courtesy Ortlinghaus GmbH

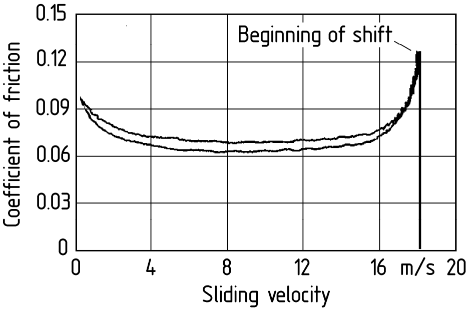


Fig. 5.99: Measured coefficients of friction for steel against sinter bronze after 400h of running in [5.85]

Wet multiple plate clutches

The main sizing criteria are the *torque capacity* and the *thermal capacity.*

The transmitted torque T (not including centrifugal pressure effects) is

$$T = z \cdot F_{ax} \cdot \mu \cdot \underbrace{\frac{2}{3} \cdot \frac{(r_a^3 - r_i^3)}{(r_a^2 - r_i^2)}}_{\text{effective radius} \approx 1/2\,(r_a + r_i)} \qquad (5.15)$$

where

z = number of friction surfaces
F_{ax} = axial actuating force
μ = friction coefficient
r_a and r_i = outer and inner friction layer radii

The simplified formula for the effective radius is ½ (r_a+r_i), which represents a very small error for multiple plate clutches. Regarding the example of Fig. 5.97, the precise value is only 0.7% larger than when calculated with ½ (r_a+r_i).

Wet multiple plate clutches usually work in transmission oil, suggesting special additives. Steel plates against layers of grooved sinter bronze are popular and robust, **Fig. 5.98**, unfortunately with low coefficients of friction, **Fig. 5.99**. The right peak represents high shear in the oil film, the left one rigid surface contact. Such a combination can withstand local temperatures of up to about 300°C.

Higher and more uniform coefficients of friction can be achieved with "paper" layers, **Fig. 5.100**; however, with slightly lower temperature limits of about 250°C. Cooling by transmission oil is usual. Contact pressures are about equal to actuating pressures.

Engaging the clutch needs a proper pressure modulation to achieve a comfortable shift, preventing layer overheating and limiting powertrain loads [5.38]. The control program has three phases, **Fig. 5.101**. The maximum torque capacity at the end is usually about 180–220% of the nominal torque. Power shifts require a certain overlapping of the pressure for the released and the newly loaded clutch [5.11, 5.38].

High torque loads and long slipping times must be handled in the case of forward-reverse "power shuttles" as the complete tractor mass must be decelerated to zero speed and again accelerated. Simulation models are recommended [5.86].

Drag losses can be a problem in the case of high speed differences of unloaded plates [5.87]. They can be minimized by

- Steel plates with *light waves*
- *Grooves* in the friction layer
- Spring type *retraction spacer*
- *Interruption of cooling flow*
- *Experimental optimization* (idling torque)

Freewheeling units can simplify a two step power shift considerably, saving one clutch and simplifying the shift control; see the famous "Torque Amplifier" of IH (1954) and MF "Multi Power" (1961).

Clamping type units, **Fig. 5.102**, are simple, but their uncertain delay for building up the torque can be a risk. This problem terminated, for example, the development of the announced Deutz Hi-Lo power shift "Multimat" based on a study of the author in 1966. Freewheeling engagement mechanics are easier to forecast for positive engagement, **Fig. 5.103**, which may be one reason that MF used a positive locking unit for the Multi Power.

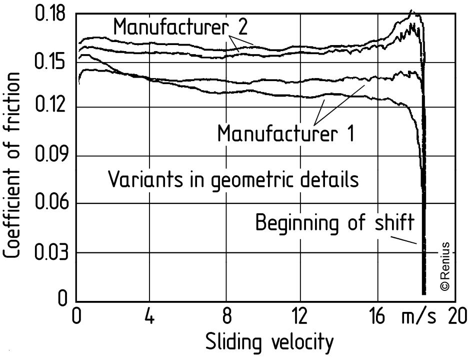


Fig. 5.100: Coefficient of friction of "paper" layers against steel plates for wet multiple clutches after 400 h of running in [5.85]

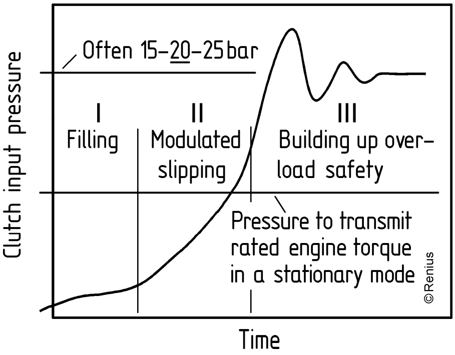


Fig. 5.101: Modulating the actuating pressure

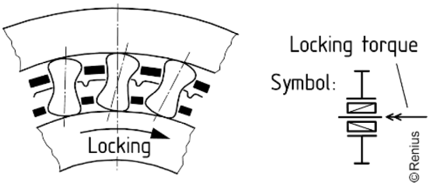


Fig. 5.102: Freewheeling unit with clamping engagement. Based on data of Ringspann

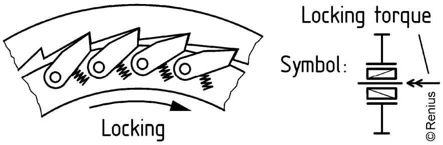
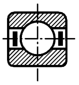
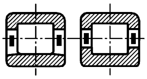
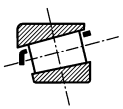
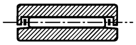
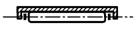


Fig. 5.103: Freewheeling unit with positive engagement

Table 5.17: Rolling bearings, guidelines

Bearing type	Design guidelines
Ball 	Universal bearing for combined radial and axial loads. Cheapest bearing for low loads. No adjustments. Tolerates certain angle errors. Expensive for high loads.
Cylindr. roller 	Cheap in the case of high radial loads, typical floating bearing. Special designs can carry limited axial loads by hydrodynamic oil films (right).
Tapered roller 	High axial and radial load capacity. Often used in pairs, needing proper adjustment. Cheap in the case of high loads. Excellent ability for stiff bearing systems of shafts, also with overhang pinions.
Needle bearings, four typical applications	
	Boring and shaft neither stiff nor hardened
	Boring and shaft stiff, but not hardened
	Boring stiff, but not hardened, shaft hardened
	Boring and shaft stiff and both hardened

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5.7.2 Transmission bearings.

Rolling (anti-friction) **bearings** are standard, journal bearings in special cases only.

Small transmissions should use ball bearings because of their low specific price at low loads, ***Table 5.17****, while cylindrical and tapered roller bearings are economic for large loads (large transmissions).*

Detailed bearing specifications are offered by the bearing manufacturers. Dimensions and nominal load capacities are standardized in ISO 281 for *defined life probabilities*, usually 90%. Axial and radial loads are combined to *equivalent radial loads.*

Tapered roller bearings are mostly used in pairs with light axial preload, **Fig. 5.104**.

Axial preload can improve the load capacity, ***Fig. 5.105****, but needs very careful adjustment and effective oil cooling.*

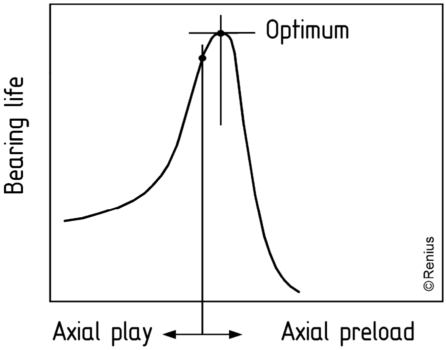

Fig. 5.105 (right)**:** Influence of axial play or preload for tapered roller bearings. Courtesy SKF

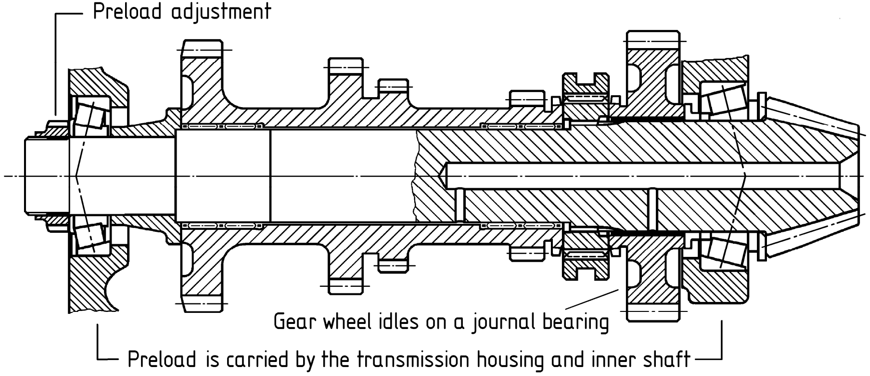
Preload adjustment
Gear wheel idles on a journal bearing
Preload is carried by the transmission housing and inner shaft

Fig. 5.104: Design of a transmission shaft with preloaded tapered roller bearings, detail of Fig. 5.29

The combination of a fixed and a floating bearing can be considered a recommended standard system, **Fig. 5.106**.

Ball bearings can carry combined axial and radial forces while cylindrical roller bearings can carry high radial forces, but in the case of Fig. 5.106 no axial forces.

Small gearboxes (for example of many Japanese tractors) are cost effective with simple ball bearings, which can also be used for preloaded pairs (Fig. 5.104) or without axial play as an alternative to the principle of fixed and floating units, **Fig. 5.107**.

The nominal load capacity C (ISO 281) is the constant radial or axial load of bearings to perform one million rotations, usually with a tolerated failure rate of 10% (B10 life L_{10}*).*

For loads F other than the nominal load C, the expected life L for all bearings is

$$L = \left(\frac{C}{F}\right)^{a} \cdot 10^{6} \text{ rotations} \qquad (5.16)$$

where a = 3 for *ball bearings* and a = 10/3 for *cylindrical roller bearings*

This equation can be regarded as the finite life fatigue leg of an S-N curve.

Failure rates lower than 10% reduce the life L being addressed by special factors.

A radial load F_r and an axial load F_{ax} (Fig. 5.106) can be combined to an equivalent radial load F:

$$F = X \cdot F_a + Y \cdot F_r \qquad (5.17)$$

The factors X and Y are listed within the specifications of the manufacturers.

If the load is not constant, the following methods of dimensioning are used:

1. The original *Palmgren* equation, published in 1924 [5.88]:

$$\frac{m_1}{n_1} + \frac{m_2}{n_2} + \frac{m_3}{n_3} + \cdots = 1 \qquad (5.18)$$

where

m_1 = bearing rotations under load 1 etc.
n_1 = bearing life (rotations) under load 1 etc.

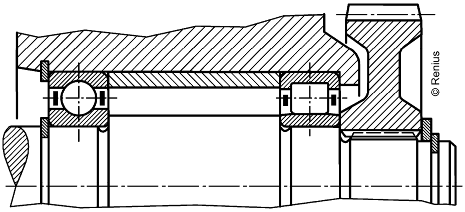
© Renius
Fixed bearing, carries radial and axial forces
Floating bearing, carries radial forces only

Fig. 5.106: Design of a shaft bearing system with a fixed and a floating rolling bearing

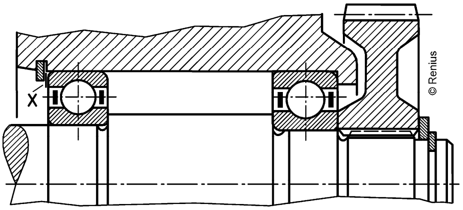
© Renius
X
Both bearings carry radial and axial forces; adjustment (x), for example, with zero axial play

Fig. 5.107: Low cost design of a shaft bearing system: Axial fixing using combined ball bearings

This “linear cumulative damage” assumption was later extended by the *Miner* rule.

2. An equivalent bearing load F_{eq} inserting load fractions 1 to i with related time fractions from a load history:

$$F_{eq} = \sqrt[a]{\frac{1}{t} \sum_{1 \ldots i} \left(F_i^{a} \cdot t_i\right)} \qquad (5.19)$$

where

a = bearing life exponent, see equation (5.16)
t = total lifespan
F_i = bearing load during time segment i
t_i = period of time of load segment i

3. Load spectrum and its direct combination with the live data of the bearing. Load spectra in time portions are suitable if the bearing speed at the main loads is relatively constant, which is mostly the case for tractors.

Otherwise speed-triggered samples are recommended [5.21].

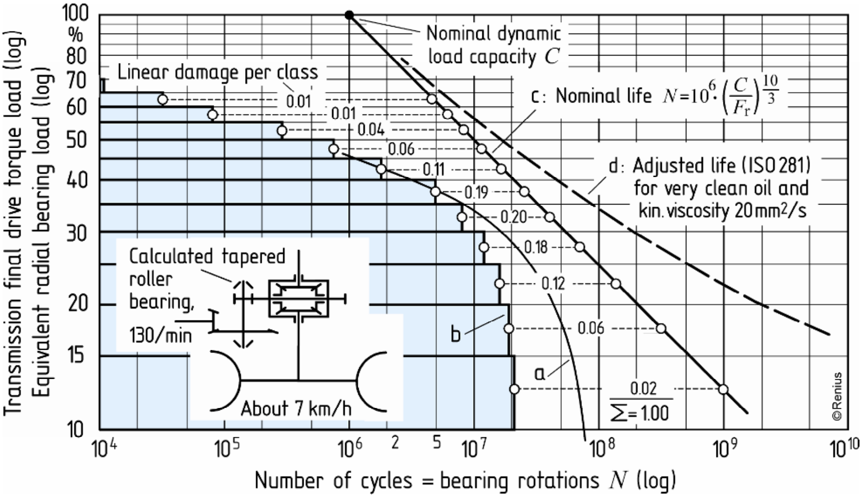

Fig. 5.108: Dimensioning of a tapered roller bearing for a given load spectrum of a tractor transmission final drive: a) sum frequency, b) class frequency [5.89]. c) standard live algorithm, d) modified rating life ISO 281. Cumulative damage calculation based on *Palmgren* [5.88]

The author has recommended method 3 since 1977 [5.89], **Fig. 5.108**. The load spectrum is usually represented in sum frequency (a) but the Palmgren-Miner rule needs a class frequency (histogram, b). The size of the bearing is acceptable if the sum of all the partial damage values is ≤1 (5.18).

Curve (c) of *nominal rating* life can be improved if the oil is very clean and the kinematic viscosity higher than the standard value resulting in *adjusted rating* life (d).

The torque load scale of this diagram offers an excellent basis for running and documenting lab tests, **Fig. 5.109** [5.89].

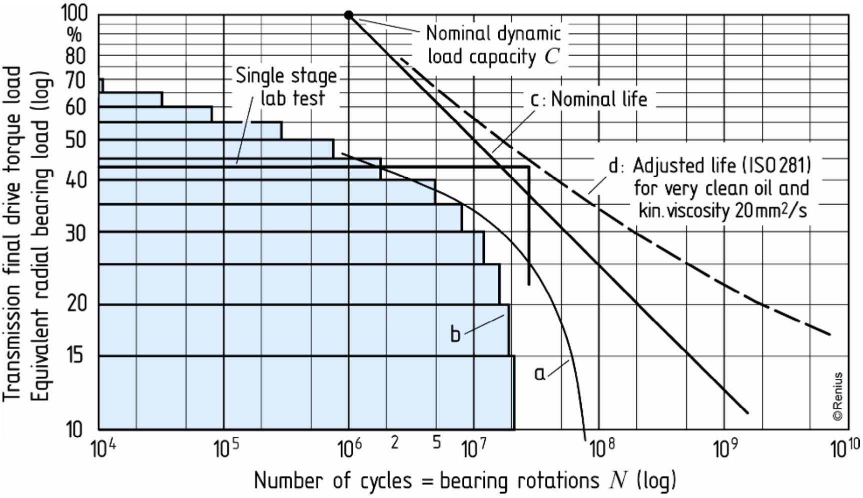

Fig. 5.109: Design verification by lab tests, starting, for example, with constant load (single stage). Probability should be documented for both, the load spectrum and the bearing life

Plain radial journal bearings can carry very high loads as demonstrated by bearings of connecting rods and crankshafts of diesel engines. Journal bearings are cheap but designing them can be challenging. Fundamentals are presented in [5.90–5.95], main parameters listed in **Table 5.18**.

A reliable journal bearing needs perfect separation of the sliding surfaces under load. This is achieved by rotation of the shaft, the bushing, or the load developing hydrodynamic pressures to carry the load.

A notch for "lubrication" in the contact area is a severe mistake, as it drains the film pressure, reducing the load capacity.

In 1902, *Richard Stribeck* (1861–1950) presented typical characteristics of friction coefficients versus shaft speeds [5.90], confirmed by *Georg Vogelpohl* (1900–75) [5.91] and others [5.92].

Plotting the friction coefficient f versus the rotational speed is not recommended as it is very difficult to keep the local viscosity constant (often ignored). Better to use the *Gümbel-Hersey number* $\eta\cdot\omega/\bar{p}$ [5.93] demanding only the *measurement* (not the control!) of viscosity η, shaft speed ω, and averaged load pressure $\bar{p}$, **Fig. 5.110**. In-depth analysis of journal bearings uses the similarity *Sommerfeld* number:

$$So = \frac{\bar{p}\cdot\psi^2}{\eta\cdot\omega} \qquad (5.20)$$

- with symbols according to Table 5.18

The position of point C depends on the relation of fluid film thickness and surface roughness [5.94]. A rough estimation of the related speed was presented in [5.95]:

$$n_{\text{ü}}\,[\text{rpm}] = \frac{F\,[\text{kp}]}{C_{\text{ü}}\cdot\eta\,[\text{cP}]\cdot Vol\,[\text{lit}]} \qquad (5.21)$$

where

$n_{\text{ü}}$ = speed of point C in Fig. 5.110 [rpm]
F = nominal journal bearing load [kp]
$C_{\text{ü}}$ = coefficient for surface quality (≈ 1–3)
η = dynamic oil viscosity [$\text{cP} = 10^{-3}\ \text{Ns/m}^2$]
Vol = volume = shaft dia. x bushing width [liter]

Table 5.18: Parameters of journal bearings

- Vertical bearing load	F
- Bore of bushing	D
- Diameter of shaft	d
- Relative bushing width	b/D
- Averaged load pressure	$\bar{p} = F/(b\cdot D)$
- Relative clearance	$\psi = (D-d)/d$
- Surface roughness of shaft	R_1
- Surface roughness of bushing	R_2
- Dynamic viscosity of fluid	η
- Angular rotational shaft speed	ω
- Materials of shaft and bushing	
- Hardness of shaft and bushing surfaces	
- Cooling flow	

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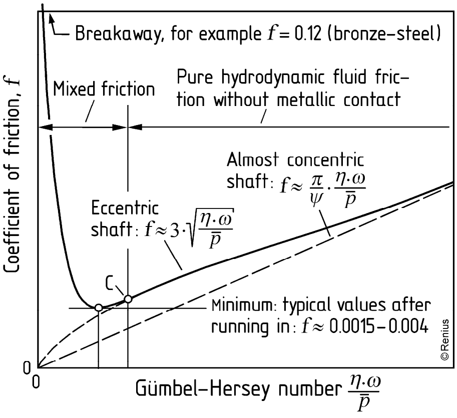

Fig. 5.110: Typical characteristic of a journal bearing, plot of the friction coefficient *f* versus the *Gümbel-Hersey* number $\eta\cdot\omega/\bar{p}$ [5.91, 5.93]. Symbols are explained in Table 5.18

The larger the $C_{\text{ü}}$, the lower (better) the $n_{\text{ü}}$. Examples of journal bearings were analyzed in [5.95], guidelines for applications in tractor transmissions are given in **Table 5.19**.

Table 5.19: Design guidelines for journal bearings in tractor transmissions for shaft diameters of ≈ 30–80mm. Symbols from Table 5.18. IT = ISO International tolerance grade

	$\bar{p}$ high, e. g. 5 MPa ω very low	$\bar{p}$ low, e. g. 0.2 MPa ω very high
ψ	0.6–1.2‰ (IT 4–5)	1.5–3‰ (IT 5–7)
η	high	low
b/d	0.5–0.8	
Bushing	bronze, fine turned	
Shaft	steel, surface hardened and ground	
Oil flow	very low	high (heat)

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5.7.3 Gear wheels

Introduction. Gear wheels are very important for the transmission costs as they are not only costly by themselves but also *influence the costs of surrounding elements such as shafts, bearings, housings, and oil fillings.*

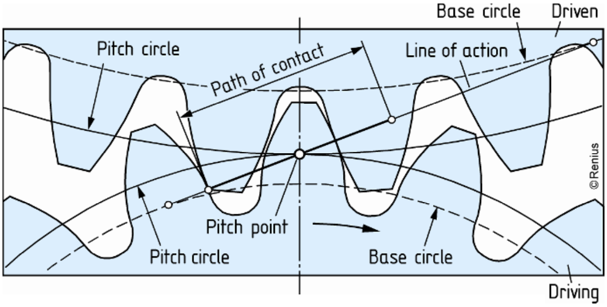

Fig. 5.111: Engaged involute gears, basic geometry

Table 5.20: Typical gear losses, % [5.96]

Gear type	Power losses load induced	Power losses by splash	Max. ratio
External spur	0.8–1.2%	0.2–1.0%	5:1
External helical	0.9–1.4%	0.2–1.0%	6:1
Spiral bevel	1.3–2.5%	0.2–1.0%	6:1
	← Speed (rpm) Ratio →	→ Speed (rpm)	©Renius

Table 5.21: Guidelines for cost-effective transmissions, as influenced by their gear wheels

- Low center distance of shafts
- Case-hardened steels such as 16MnCr5, 20MnCr5,14CrNi6, 18CrNi8, 18CrNiMo7-6. Case hardening depth about 15% of module Surface hardness 58-63 HRC
- For planetary ring gears also 42CrMo4
- All gear wheels based on forged green bodies
- Helical gears (15–25°, performance, noise), direction and value of angles on one shaft should compensate for axial bearing loads
- Shot peening for increased bending strength
- Positive rack shift at the pinion (performance)
- Nominal pressure angle (tool) 20–25 (28)°
- Tooth root radius as large as possible
- ISO quality 6–8 (ground after hardening)
- Geometric corrections in the cutting process for compensated deflections under load
- The four basic S-N curves (in load scale!) adjusted to the load spectrum (Fig. 5.112)

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Table 5.22: The four basic criteria of dimensioning a two-gear system of a transmission

- Pinion tooth bending
- Pinion tooth surface stress
- Wheel tooth bending
- Wheel tooth surface stress

Involute profiles are the standard geometry, **Fig. 5.111**.

They are easy to define and to generate by a reference tooth rack with linear flanks and they offer a favourable tolerance against center distance errors. The point of contact moves on a straight line which is a tangent on both base circles.

The transfer of the circumferential force can be understood as a slipless rolling of two cylinders, each with the pitch circle diameter. In reality there is pure rolling only at the pitch point, with negative and positive specific sliding aside it during the contact.

However, it is mainly the partial rolling which creates extremely high elasto-hydrodynamic fluid pressures of up to the very high value of about 2500 MPa.

The oil film generation is supported by a huge increase of the dynamic viscosity due to the very high pressure; the partial rolling explains the low losses, **Table 5.20** [5.96].

Tractor transmissions have a high power density, **Table 5.21** gives recommendations for the design of cost-effective gears.

Case-hardened high-grade steels are standard materials for gears of tractor transmissions, offering best possible strength for tooth bending and flank surface pressure.

Two gears represent a gear system and must always be designed together. There are at least four criteria of dimensioning important for transmissions, **Table 5.22**.

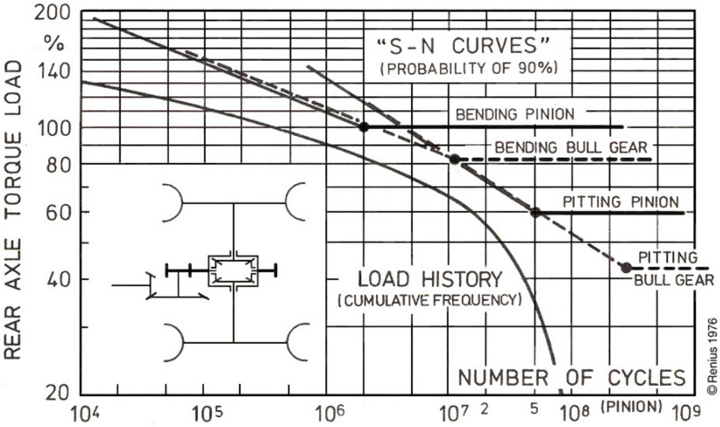

Fig. 5.112: Cumulative damage diagram for a pair of gear wheels demonstrating optimized dimensioning by S-N curves, scaled in axle torque, coping with a given torque load spectrum. Cumulative damage sum (Palmgren-Miner-Haibach) = 1. Original diagram of *Renius* 1976 [5.23]

Dimensioning of gear wheels. Several interesting publications have been presented to find the most economic specifications. If they are based on constant nominal loads, an optimum is usually not possible in the case of tractor transmissions as they are subjected to a very huge bandwidth of torque loads. Assuming constant loads can result in confusing safety factors (shown in Table 5.26 below). This situation motivated the author around 1968 to start basic investigations at TU Braunschweig along with analyzing the gear wheels of several mass-produced tractor transmissions.

After being employed in 1973 by KHD (Deutz), he developed an optimization as presented first in 1976 in the US [5.23] and in 1977 in Germany [5.89], **Fig. 5.112**.

The diagram shows optimal "S-N curves" for a given load spectrum. It is scaled vertically in torque (not in stress) in order to demonstrate the gradation and to simplify lab tests. Therefore, the "S-N curves" could also be termed "T-N curves". The horizontal scale represents the number of cycles (engagements), normalized for the pinion. This allows keeping the load spectrum fixed. Each "S-N" curve represents an ideal position, found by applying the cumulative damage calculation of *Palmgren-Miner-Haibach* [5.89]. The beginning of infinite life for bending was originally set at 2 x 10^6 cycles and for pitting at 5 x 10^7 cycles.
ISO 6336-3 now recommends an infinite life of bending from 3 x 10^6 cycles.

The benefit of this strategy is that the optimal "S-N" curves are graded, offering potential for cost savings for three of the four basic criteria, with the largest effect for pitting of the bull gear (see also chapter 8.2).

The statistical reliability of the procedure can be influenced by the probability of the load spectrum and of the S-N curves (90–95% recommended for both). Additional safety is possible by reducing the damage sum, for example for bending from 1.0 to 0.5.

The "Palmgren-Miner-Haibach" rule was proved by research projects at TU Munich [5.97–5.100]. *The author states from long term experience that the main uncertainty is not the damage model, but the forecast of (load-based) S-N curves. Hence, gathering experience is highly recommended.*

General fundamentals see [5.102, 5.103].
Important standards: AGMA 2001-D04, DIN 3990, and the ISO family ISO 6336.

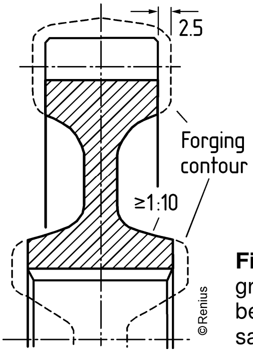

Fig. 5.113: A forged green body increases bending strength and saves material

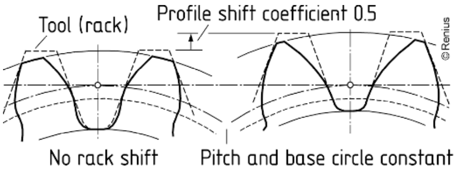

Fig. 5.114: Influence of a positive rack shift of 0.5 x module on the tooth shape. Pinion with 12 teeth. Very common to increase tooth stability

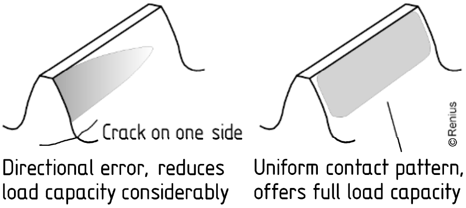
Crack on one side
Directional error, reduces load capacity considerably
Uniform contact pattern, offers full load capacity
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Fig. 5.115: Misalignment of teeth

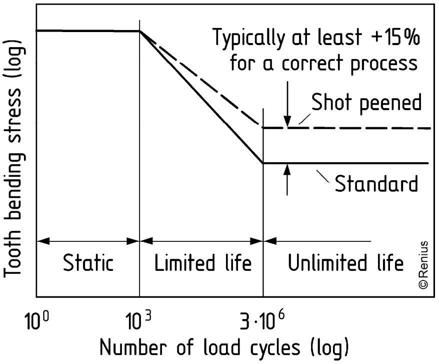


Fig. 5.116: Increased bending endurance limit for gears of carburized steels by shot peening. Load cycle sections according to ISO 6336-3

Factors to be applied for achieving the optimal "S-N" curves. The strategy is based on Fig 5.112. The infinite life load level of the four "S-N" lines (in load scale) are influenced by the following parameters:

- *Module:* Large values usually favour bending strength, reduce pitting resistance. Module = ref. circle dia. / number of teeth
- *Material:* Cr-Ni steels offer slightly higher bending endurance limits than 16MnCr5 or even 20MnCr5.
- *Forging* (green body) increases bending strength (versus blank), **Fig. 5.113**.
- *Heat treating*: The higher the surface hardness, the higher the pitting resistance. Usual values for hardness: 58-63 HRC.
- *Rack shift coefficient:* Positive values (retracted rack tool) increase bending strength due to a larger tooth base. The lower the number of teeth, the higher the influence on the shape, thus being very important for pinions, **Fig. 5.114**.
- *Correct contact pattern:* Misalignment of the teeth, **Fig. 5.115**, reduces the endurance limit for bending and pitting dramatically. The cause is often an elastic deformation which can be compensated for by a special cutting or grinding process.
- *Pressure angle:* Increased values (above 20°) can improve bending strength.
- *Surface quality:* Low values of surface roughness increase pitting resistance.
- *Oil:* High viscosity and adequate additives increase pitting resistance; however, high viscosity also increases splash losses.
- *Shot peening* notably increases the bending endurance limit, **Fig. 5.116** [5.104], reducing the spread at the same time [5.105].

All these listed influences can be used to optimize the "S-N" curves in the sense of Fig. 5.112, *saving costs by their graded position.* An early systematic application was the Deutz tractor transmission TW90 [5.106], successfully introduced and produced in about 150,000 units.

Calculation of bending stress. The maximum bending stress is generated when the tooth force acts on the outer point of single tooth contact. The compressive stress (on the left side, not shown) is slightly higher than the tensile stress on the right side because of the radial force; however, cracks are usually initiated by the right tensile stress, **Fig. 5.117**. The calculation is standardized in ISO 6336-3:

$$\sigma_F = \underset{\text{Tooth form}}{Y_F} \cdot \underset{\text{Stress correction}}{Y_S} \cdot \underbrace{\cdots\cdots\cdots}_{\text{Further factors}} \cdot \frac{F_t}{b \cdot m_n} \tag{5.22}$$

where

F_t = nom. force = torque / reference radius
b = effective gear face width
m_n = normal module

The factor Y_F mainly considers the conversion of the basic bending mechanics to the use of the expression $F_T/\mathrm{b} \cdot \mathrm{m}_n$, and Y_S considers the notching effect due to ρ_F.

Y_S depends very sensitively on the tooth root radius ρ_F. It is highly recommended that Y_S is kept as low as possible [5.107] *by defining the cutting tool for the largest possible radius ρ_F* (computer simulation).

Special cases. If gear teeth are loaded *cyclically from both sides* (e. g. in a planetary gearing) with the same load and service time, the bending stress endurance limit must be reduced by a factor of 0.70 (ISO 6336-5) or 0.75 [5.108].

Another special case is a low *rim thickness*. The guideline of **Fig. 5.118** [5.109] stands for full bending capacity.

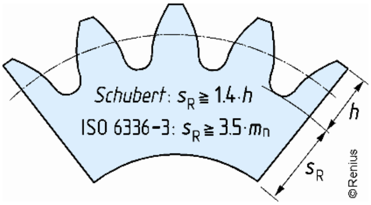

Schubert: $s_R \geqq 1.4 \cdot h$
ISO 6336-3: $s_R \geqq 3.5 \cdot m_n$
h
s_R
©Renius

Fig. 5.118: Necessary rim thickness for full bending capacity without a reduction factor [5.109]

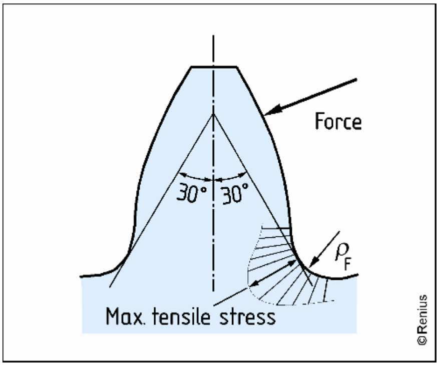


Fig. 5.117: Geometry and tooth bending. Note the stress increase as a result of the notching effect

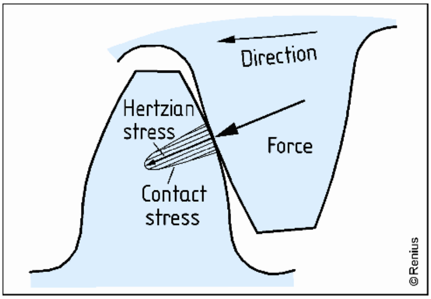


Fig. 5.119: Contact stress and Hertzian stress of an elasto-hydrodynamic gear contact

Calculation of contact stress. The contact force creates contact stress, **Fig. 5.119**, with a maximum which is responsible for fatigue, resulting in *pittings* when the fatigue limit is exceeded.

The Hertzian stress calculation is standardized in ISO 6336-2 resulting in

$$\sigma_H = \underbrace{\cdots\cdots}_{\text{Five factors}} \sqrt{\underbrace{\cdots\cdot}_{\text{Four factors}}} \; \sqrt{\frac{F_t}{b \cdot d_1} \cdot \frac{u+1}{u}} \tag{5.23}$$

where

F_t = nom. force = torque / reference radius
b = gear width
d_1 = reference diameter of pinion
u = gear ratio (out/in), neg. for internal gears

Note, that σ_H is not proportional to nom. load F_t but proportional to its root $F_t^{1/2}$!

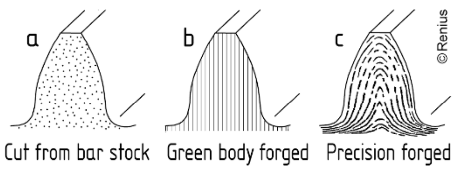


Fig. 5.120: Material structure (polished cut) of gear wheels for three production processes

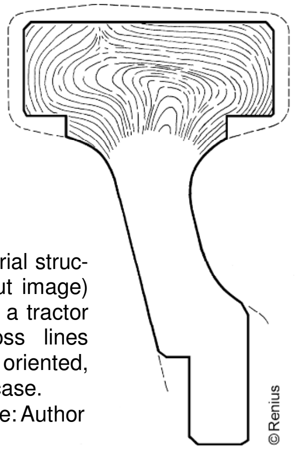


Fig. 5.121: Material structure (polished cut image) of a bull gear of a tractor final drive. Dross lines should be radial oriented, not ideal in this case.
Source: Author

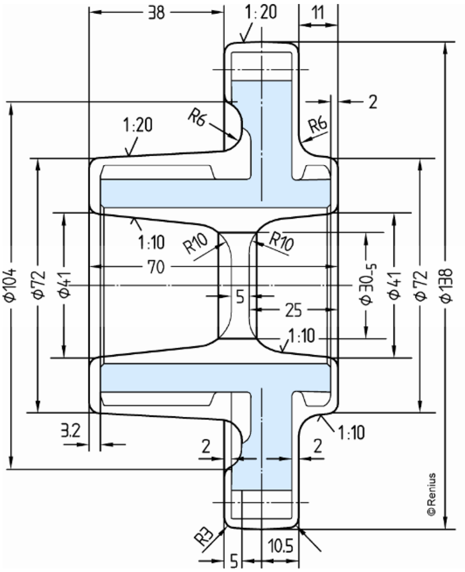


Fig. 5.122: Example of a forged gear green body. Courtesy KHD (Deutz)

Calculation of "S-N" curves. Horizontal legs are found when their level corresponds to safety factor 1.0 and life factor 1.0. Equations (5.22) and (5.23) are used to find the related nominal loads F_t. The endurance limit inflexion points (Fig. 5.112) require the following values:

- Endurance limits for σ_F and σ_H
- Inflexion point pitting: $5 \cdot 10^7$ cycles (ISO)
- Inflexion point bending: $3 \cdot 10^6$ cycles (ISO)

A similar procedure is carried out for the inflexion points for the static load limits: 10^4 cycles for pitting (ISO 6336-2) and 10^3 for bending (ISO 6336-3). The calculated tangential loads are converted to torques. The results allow plotting of the four "S-N" curves (in log-log scale, Fig. 5.112).

Forging of green bodies (Fig. 5.113). This is a very common method for vehicle transmissions including tractors. Forged green bodies are the first step of producing gear wheels with the following advantages:

- Material savings
- Machining effort and energy savings
- Higher material strength, mainly for tooth bending

Two effects improve the strength:

The first one is the influence of the structure of dross lines, **Fig. 5.120** and **5.121**. These lines should have a radial direction in order to get high bending stress limits.

The second effect is a densification of the material initiating preloaded stresses.

It may be noted that the bevel planets in the differentials of many vehicle axles are usually only forged, Fig. 5.120, Version c.

The structure of Fig. 5.121 is not very satisfying (no perfect radial lines). If a reinforcement is required, a vertical exaggeration of the forged green body in the top area could improve the radial orientation.

Forged green bodies need a drawing for machining the forging tool as demonstrated by **Fig. 5.122** for a realized example.

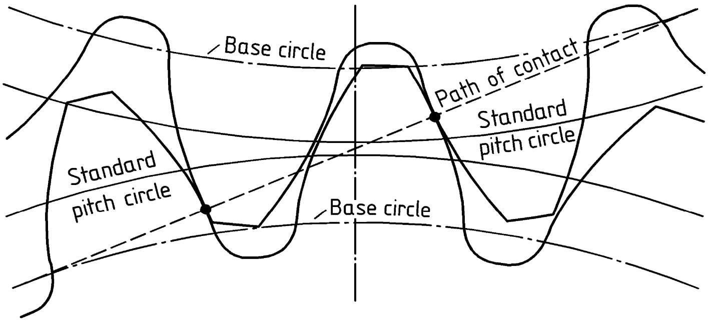
Example of specifications of a gear set. In order to give an impression of the typical data and their complexity, they are listed as an example in **Table 5.23** which may also help to test computer programs. They are highly recommended for completing basic data and plotting the final geometry for visual validation.

Table 5.23 is an output extract of the program StPlus, which was developed by the Gear Research Center FZG of TU Munich in co-operation with FVA (German Research Association of Drive Systems). The gear set is used for a laboratory test rig. The grade of accuracy can be slightly lower for tractor transmissions

Table 5.23: Example of specifications and geometry of a gear set. Data and original geometry plot courtesy *K. Stahl* and *M. Hain*, FZG TUM Munich 2017

	Pinion	Gear wheel
Number of teeth	22	24
Center distance, mm	112.500	
Face width, mm	55.0	
Material of both	18CrNiMo7-6	
Heat treatment	case carburized	
Surface hardness, HV (HRC)	750 (62.2)	750 (62.2)
Helix angle at pitch diameter, deg	29.0000	
Grade of accuracy (DIN 3961)	N 5	N 5
Tool normal module, mm	4.25	
Transverse module, mm	4.85925	
Tool pressure angle, deg	20.0000	
Working transv. press. angle, deg	23.48027	
Sum of profile shift coefficients	0.1767	
Profile shift coefficient	0.1000	0.0767
Normal circular pitch, mm	13.352	
Length of path of contact, mm	21.195	
Transverse contact ratio	1.504	
Overlap contact ratio	1.997	
Total contact ratio	3.501	
Base diameter, mm	98.698	107.671
Working pitch diameter, mm	107.609	117.391
Standard pitch diameter, mm	106.904	116.622
Radius coeff. of tool tip	0.249	0.249
Generating profile shift	0.1757	0.1524

Geometry plot:



5.7.4 Planetary gear systems

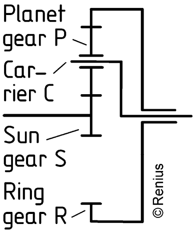


Fig. 5.123: Configuration of the three-shaft standard planetary, example for three planets

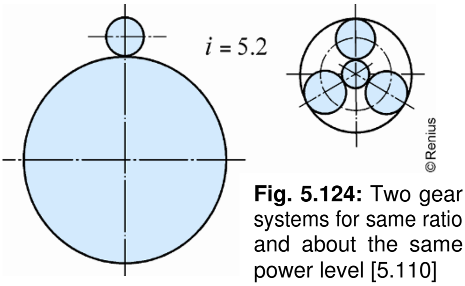


Fig. 5.124: Two gear systems for same ratio and about the same power level [5.110]

The most-used configuration is shown in **Fig. 5.123**, very important for tractor final drives, sometimes used for gearbox range shifts (MF 65) and power split CVTs (ZF).

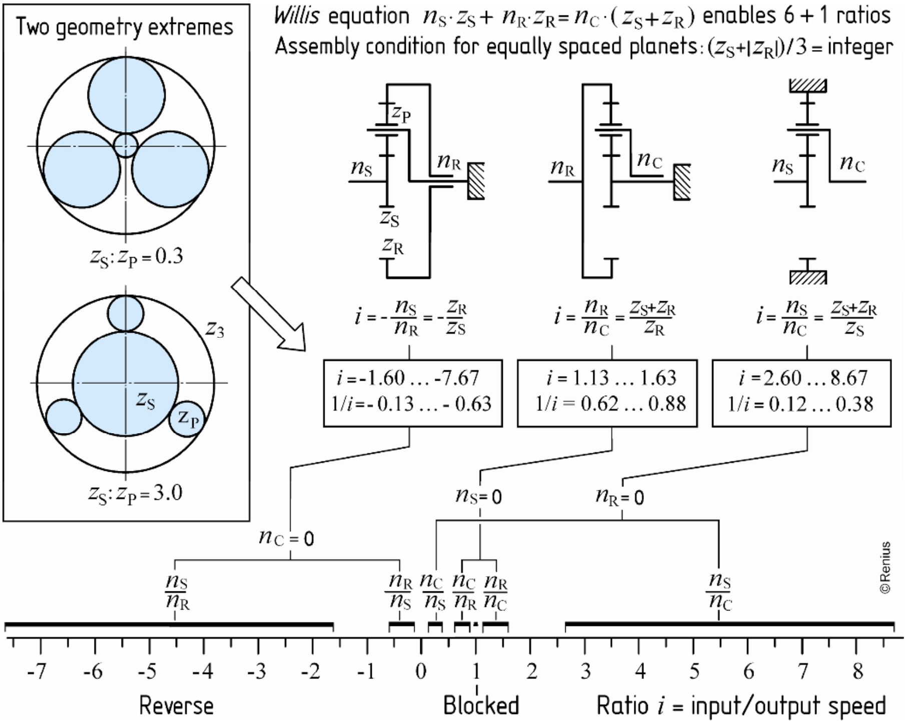
It offers outstanding power density due to the power split between sun and planets, **Fig. 5.124**.

In the case of three planets, the sun gear is self-centring if $(z_S + z_R)/3$ equals an integer (z_S and z_R see **Table 5.24**). *This spares a bearing and delivers a perfect torque distribution.* Four planets are usually not able to increase the torque capacity linearly by a factor of 4/3 but, for example, by 20-25%.

The seven potential ratios for the standard planetary are demonstrated in Table 5.24.

The ratio of n_S / n_C is very important for tractor final drives. The upper limit of a about 9:1 is very cost-efficient but requires high skills in design and testing.

Table 5.24: Standard planetary system, practical ratios. z = number of teeth, n = rotational speeds



Example of a power reverser planetary structure. *A forward-reverse power shift is difficult with the standard planetary, as a reasonable ratio of about -1.0 to -0.9 for tractors is not possible, Table 5.24.*

A recommended solution is the structure of **Fig. 5.125** which was, for example, applied by John Deere in several transmissions with the advantage of a compact installation space.

The forward ratio is 1:1 (clutch closed, system blocked). Calculation of the reverse ratio is possible with the *Willis* equation.

Robert Willis (1800–1875) was a British engineer; his model for three-shaft planetary systems is astonishingly simple and universal and shall be applied to this case as an example:

$$n_S - i_0 \cdot n_R - (1 - i_0) \cdot n_C = 0 \qquad (5.24)$$

where

n_S = rotational speed of sun gear
i_0 = stationary ratio (at carrier speed $n_C = 0$)
n_R = rotational speed of ring gear
n_C = rotational speed of carrier

The value for i_0 is in this case:

$i_0 = z_R / z_S$ with z for the number of teeth.

The actual ratio in reverse mode can now be calculated by inserting $n_R = 0$ (brake engaged), $i_0 = z_R / z_S$ and z_R positive in equation (5.24) which results in:

$$\frac{n_C}{n_S} = \frac{1}{1 - z_R / z_S} \qquad (5.25)$$

The numbers of teeth are proportional to gear wheel diameters. This means that the diameters of Fig. 5.125 can be used for estimating the reverse ratio; in this case, about 0.9 from equation (5.24). This means that the reverse speeds are about 11% faster than the forward speeds. Such a value meets typical market demands.

John Deere has produced this power reverser in very high volumes, mainly for tractors of the 6000 line since its introduction in 1992 (Fig. 5.32).

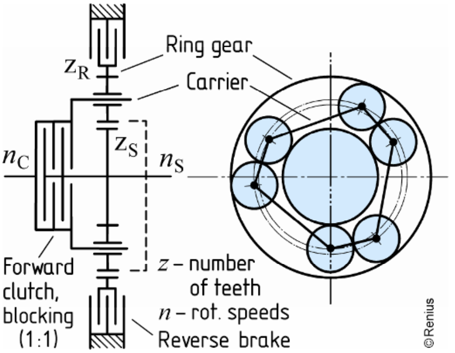


Fig. 5.125: Planetary system for a power reverser. Clutch: Forward; brake: Reverse. Applications see, for example, John Deere 4-speed power shift and John Deere CVTs

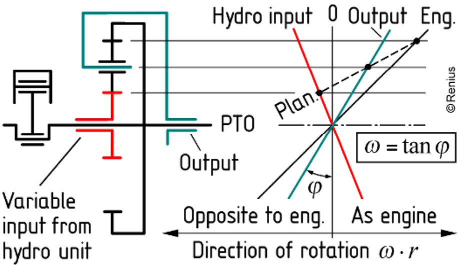


Fig. 5.126: Analysis of the standard planetary with the "Kutzbach" speed chart [5.111]. Example: Graphic solution for the carrier speed of the merging planetary of ZF CVT, Fig. 5.71

The **"*Kutzbach* chart"** is recommended to demonstrate the internal movements within a planetary, mainly for the case that all shafts are in rotation.

As an example, **Fig. 5.126** shows the merging planetary of the ZF CVT of Fig. 5.71.

If the engine speed and the hydro unit input speed are given, the dotted line of the planetary wheel can be drawn. The key for doing this is the fact that the planetary wheel has the same circumferential speed as the sun gear at their inner contact and the same circumferential speed as the ring gear at their outer contact. Between these contacts, speed distribution of the planets is linear; interception leads to output speed.

Calculation of cycles for standard planetary gear wheels. Cycles are needed for fatigue dimensioning, **Table 5.25**.

The sun gear has the highest number of contacts and usually needs the most intensive efforts to achieve adequate life.

Table 5.25: Number of engagements for the three gear wheels of a standard planetary (Fig. 5.123) for one rotation of the carrier

Total number of contacts of sun gear for one carrier rotation	$N_S = p \cdot \frac{\lvert z_R \rvert}{z_S}$
Contacts of planetary gear (with sun gear) for one carrier rotation	$N_{P\text{-}S} = \frac{\lvert z_R \rvert}{z_P}$
Contacts of planetary gear (with ring gear) for one carrier rotation	$N_{P\text{-}R} = \frac{\lvert z_R \rvert}{z_P}$
Number of contacts of the ring gear for one carrier rotation	$N_R = p$

Notation: p = number of planets, other symbols see Table 5.24

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The number of planets *p* is often three, with the important advantage that the sun gear is well self-centring with no need for a second bearing, **Fig. 5.127**.

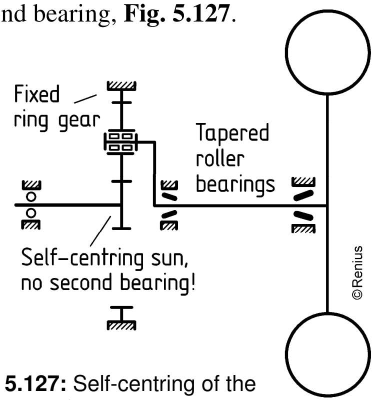


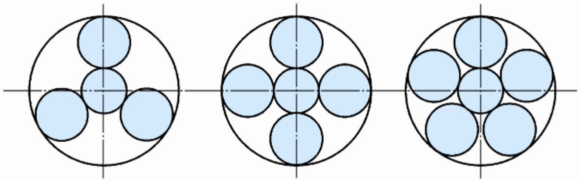
Fig. 5.127: Self-centring of the sun gear of a rear axle planetary with three planets

The torque capacity of the standard planetary can be increased by using more than three planets, **Table 5.26.**

However, the additional performance gain is slightly below ideal conditions and the maximum possible reduction ratio is lower than with three planets. Because of these reasons, tractor rear axles usually have three planets; front axles sometimes four (ZF) and very seldom five.

The teeth of the sun gear must be cut with relatively large corrections in order to compensate for the high torsion deflection because of the multi-contacts and its usually small diameter. This requires special skills in design with FEM modelling and final tests. The sun gear also needs excellent materials and treatments.

Table 5.26: More than three planets increases torque capacity but decreases max. reduction ratio



Torque capacity, ideal	100%	133%	167%
Planetary loads	uniform	almost uniform	almost uniform
Torque capacity, realistic	100%	about 125%	about 155%
Maximum reduction ratio	very high	high	moderate

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5.7.5 Rear axles

The complete rear axle, which usually contains the brakes and the PTO gearing, represents about 40% of the transaxle costs.

Regarding tractors of the upper technology levels with 4WD, this results in about 12% of the total initial tractor costs.

Fundamentals for brakes and PTOs have already been presented in previous chapters.

The main components of the final drive are:

- Spiral bevel pinion and large crown gear
- Differential with bevel planetary gears
- Differential lock
- Final drive

Table 5.27 contains basic structures.

Design A was used over many years for Deutz tractors together with the gearbox TW 50 (Fig. 5.12) and is still of interest for low-cost tractors below about 40 kW.

For many decades **Design B** was popular, typically for higher power levels; however, it was more or less replaced by **design C** which is now dominating.

C offers higher power density and higher potential of total axle ratio (Fig. 5.124, Table 5.24); both are needed for axles of increased power levels and tire diameters.

A differential ratio of about 5 seems to be reasonable regarding the stresses of the pinion and its minimum number of teeth.

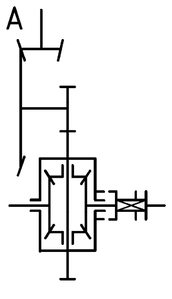
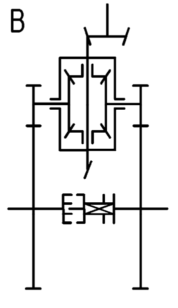
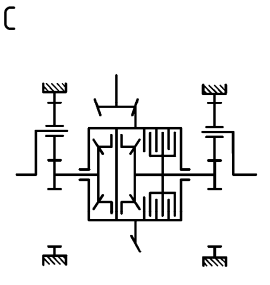
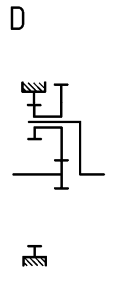
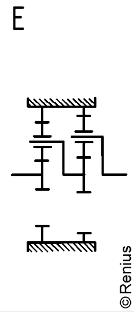
A typical speed reduction ratio of the total axle of structure C for a 200 kW tractor could be 5:1 for the bevel gearing and 8:1 for the planetaries: A total of 40:1.

High-powered tractors with tire diameters above 2 m can need even more. One possibility is to apply more complex planetary structures, such as in **designs D** and **E**.

This is not a functional problem but results in *higher specific initial costs*, which is the main reason for making the very best of structure C. As such a rear axle planetary can be a bottleneck for the power throughput, its design requires outstanding skills in the development process. An important principle is, for example, to give the pinion shaft freedom for self-centring by the planet gears (Fig. 5.127). This requires that the teeth installation condition is fulfilled (Table 5.24). A second important principle is *precise geometric compensation of the high elastic pinion twist angle.*

Designs B and C are presented in two models, **Fig. 5.128** (Deutz) and **Fig. 5.129** (J. Deere), both produced in large series.

Table 5.27: Structures of rear axle final drives. The applied power levels increase from left to right

A	B	C	D	E
				
Tech level: I-II	II-III	(II)* III-V	IV-V	
Ratio: 20-25	20-30	30-45	>45 (with diff.)	

* With simple differential lock as common for A and B

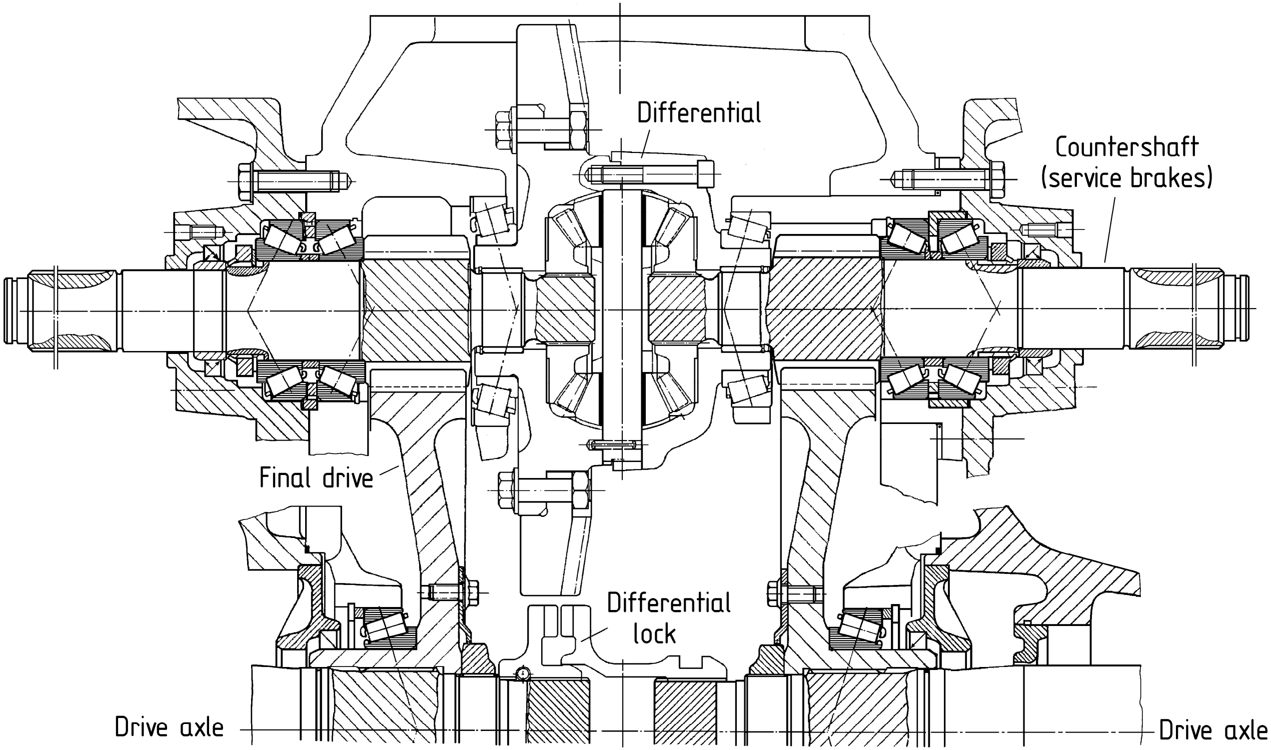


Fig. 5.128: Final drive type B (Table 5.27), Deutz transmission TW 90 (1978–82) and TW 900 (1982–91, now 40 km/h top speed). Largest model (about 80 kW). Two pairs of helical spur gears, 10/59 teeth, center distance 252.5 mm. Bull gears in overhang position, deflections carefully compensated by gear geometries. 4800 kg net weight (with cab and all-wheel drive). Maximum nominal weight 6800 kg, maximum rear tires 18.4R38 (radius 0.82 m). Special benefit: 50:50 torque split at the differential when it is locked, reduces the load spectra of the 10/59 gears and of the final drive shafts as compared with structure C.

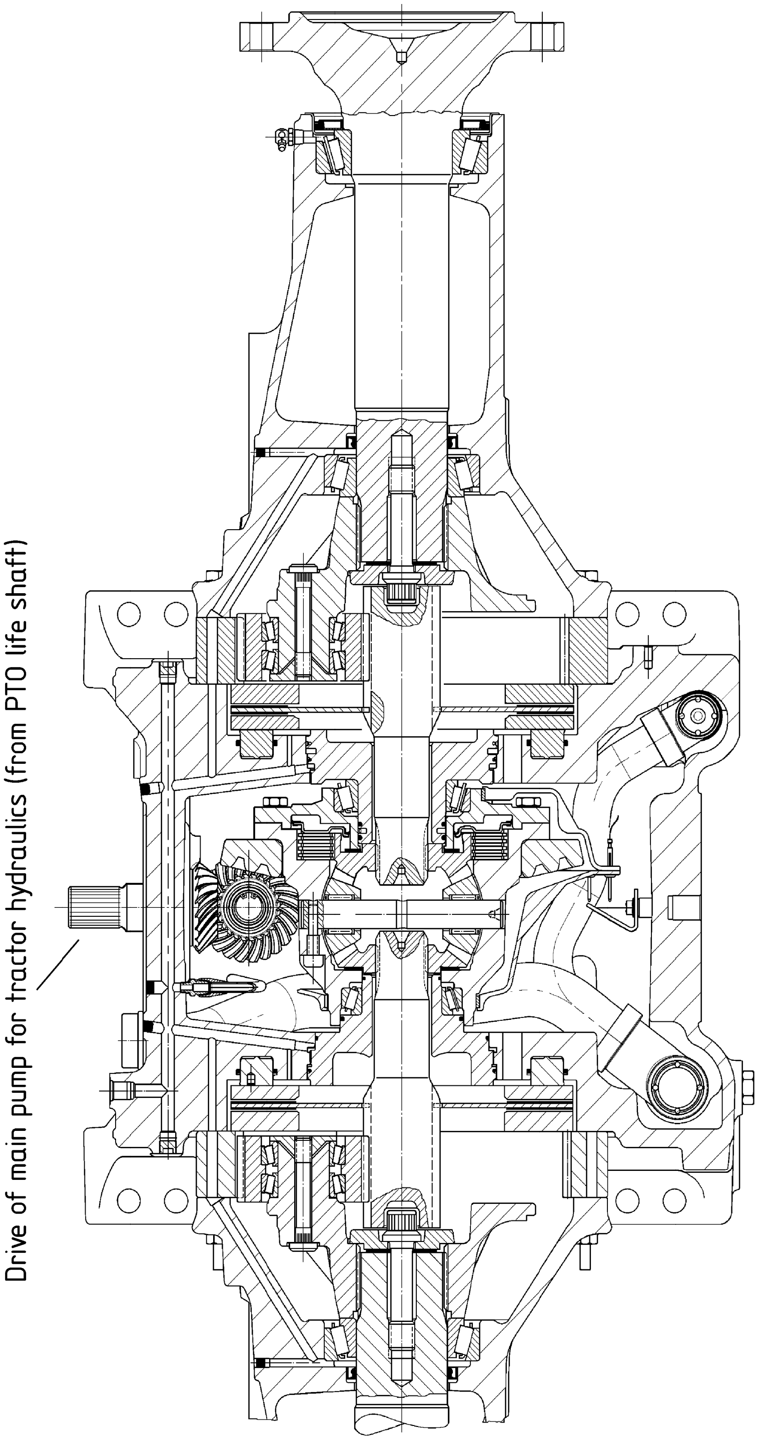

Drive of main pump for tractor hydraulics (from PTO life shaft)

Fig. 5.129: Rear axle of the John Deere tractors 6145R and 6155R, maximum rated engine power about 120 kW, maximum tire radius about 0.9 m. Planetary ratio 6.4. Wet disc brakes. Differential lock hydraulically operated.
Drawing based on CAD material of John Deere, kindly released by John Deere Werke Mannheim 2017

5.8 Strategies of dimensioning

5.8.1 Tractor transmission loads

Tractor transmission loads are not uniform, but irregular with the highest peaks at abrupt clutching in gears of higher speeds, **Fig. 5.130** and **Fig. 5.131**. Maximum values can be equivalent to those at about four times the nominal engine torque, due to dynamic mass forces of the clutch plate even above the nominal slipping torque.

However, these torques occur over only a very short time. In contrast, the average drive transmission load is only about 40–50% of the nominal torque. This is about *ten times lower* than the peaks.

This complex relationship requires statistical load evaluation methods.

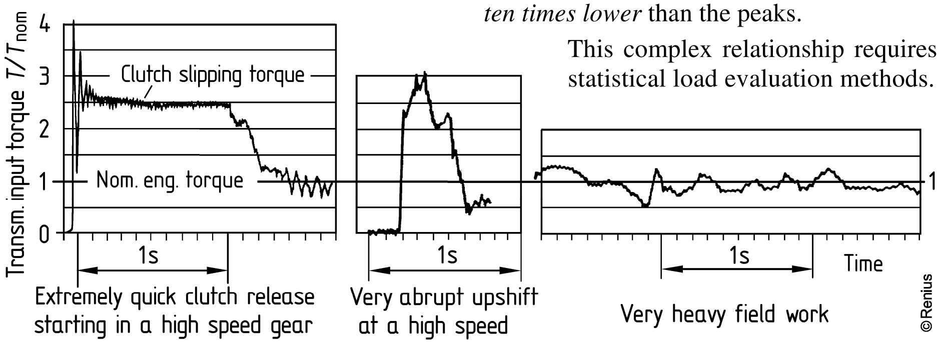


Fig. 5.130: Record of measured loads at the transmission input shaft for three typical cases [5.112]

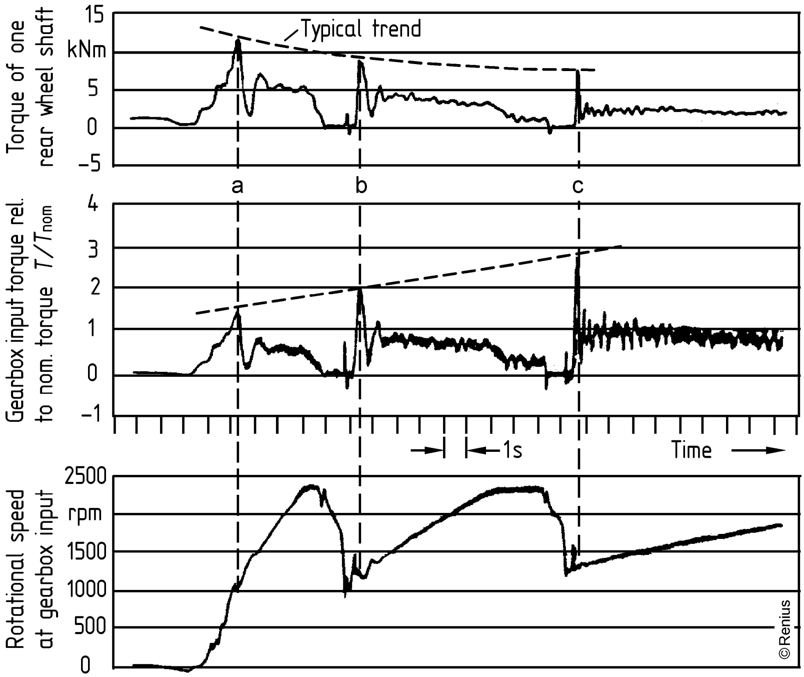


Fig. 5.131: Typical measured powertrain loads and speeds during upshift. Tractor 50 kW, 4t, pulling a 10t trailer. Nominal gear speeds: a) 9 km/h (starting), b) 14 km/h, c) 18 km/h [5.21]

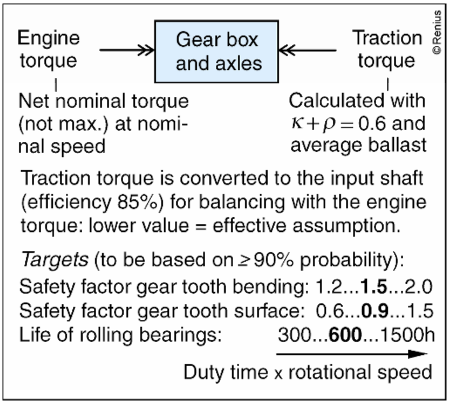
5.8.2 Simplified constant loads

A first step of dimensioning transmissions uses constant loads, **Table 5.28.** Input and output torque must be balanced if we assume stationary conditions. It is recommended to convert the traction torque with the transmission ratio and efficiency to the input shaft; the lower value of both is valid.

Rule: Transmission elements of the lower tractor speeds (up to about 5 – 7 km/h) are calculated with the traction torque and elements of higher speeds with the engine torque.

The recommended safety factors (related to endurance limit) represent no real safety. This contradiction can be resolved well by cumulative damage analysis (Fig. 5.112).

Table 5.28: Strategy with static loads. Stepped transmissions, dry clutch, 10,000 h life, torque back-up ≤15%; κ and ρ defined in chapter 2.2. Targets slightly higher for higher torque back-up



Engine torque → Gear box and axles ← Traction torque

Net nominal torque (not max.) at nominal speed

Calculated with $\kappa + \rho = 0.6$ and average ballast

Traction torque is converted to the input shaft (efficiency 85%) for balancing with the engine torque: lower value = effective assumption.

Targets (to be based on ≥90% probability):

Safety factor gear tooth bending: 1.2...**1.5**...2.0

Safety factor gear tooth surface: 0.6...**0.9**...1.5

Life of rolling bearings: 300...**600**...1500h

Duty time x rotational speed →

5.8.3 Cumulative damage analysis

The first step is to have a **load spectrum**, **Fig. 5.132**.

Rule: The statistical format of a load spectrum must be adapted to the case of application.

Rainflow counting is an excellent method for analyzing stress histories in structures.

Load spectra from rainflow counting are, however, not suitable for dimensioning gear wheels, rolling bearings, and shafts (shafts under rotating bending moments).

These elements always have one internal stress cycle per engagement! *Hence, the load determines their stress amplitude; the speed, the frequency!*

Consequently, the best procedure for processing measured load tracks for dimensioning these elements is to record load samples triggered by the rotational speed (also recommended in ISO 6336).

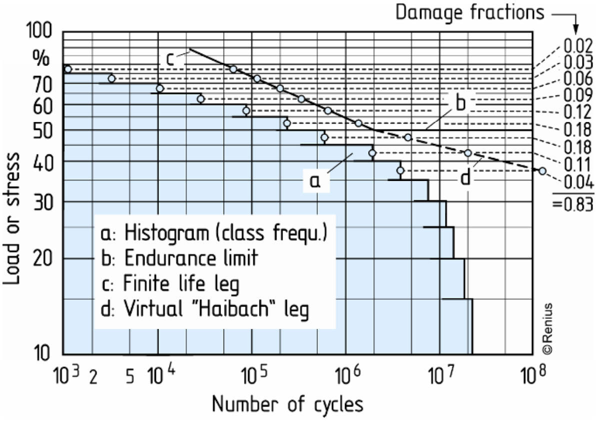


Fig. 5.132: Damage calculation after *Palmgren-Miner-Haibach.* Example from 1977 [5.89]: Gear wheel, tooth bending

Rule: *A constant speed results in time portions.*

Life evaluation is done by linear cumulative damage calculation, Fig. 5.132. The given load spectrum in *class frequency* is defined for the total life of, for example, a gear wheel. The S-N curve b-c is calculated in *load for the mounted gear.* The limited life leg c is prolonged by the “Haibach” leg (d) with reduced slope exponent - (2*k*-1) instead of - *k* [5.113]. The damage fraction is the ratio of actual class cycle and cycle to failure. The sum of all damage fractions must be less than 1 for the given life.

In the past, transmission durability tests were often carried out in the field. Field tests are still used, but lab tests are cheaper, faster, and repeatable, thus more important today.

In a first phase of introducing lab tests, the engineers involved often developed their test specifications independently of the design department.

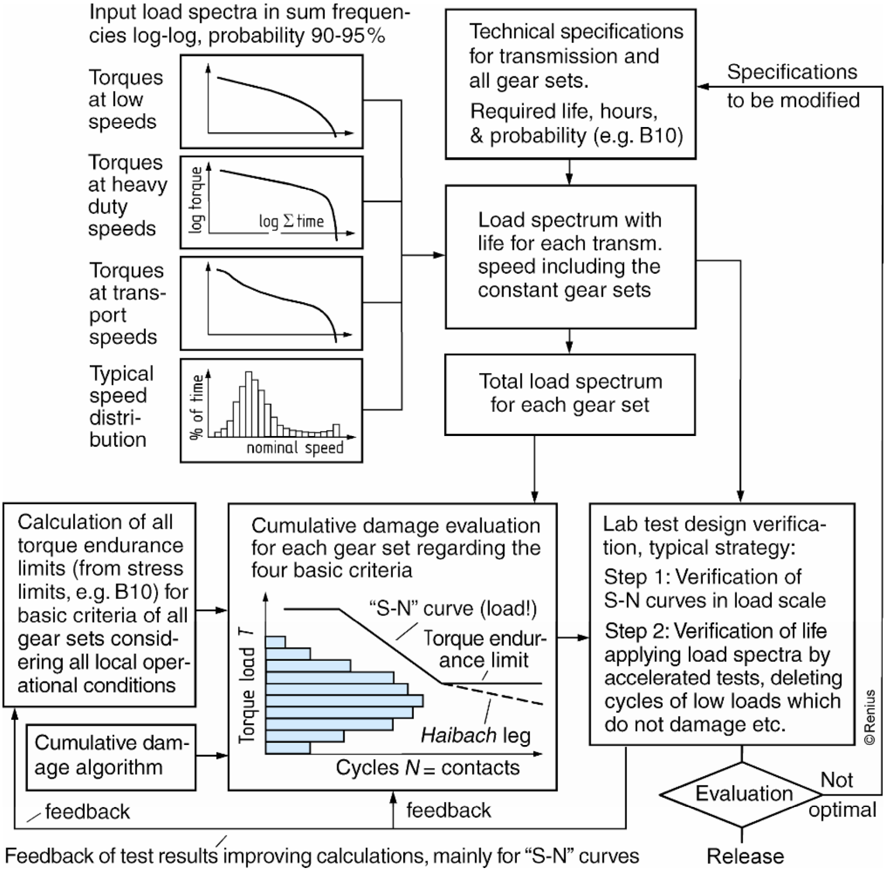
Cumulative damage analysis allows integrating both, modelling and testing in one system, offering the advantage of better test result feedback for improved modelling, **Table 5.29.** This system has been proved by the author in many projects since 1979.

The first step is to define realistic load spectra for a given probability, often 90 or 95% (B10 or B5). Typical standard load spectra are given below. Speed spectra see chapter 5.2, PTO load spectra chapter 5.4.

Measurements are carried out for individual agricultural operations. For the final drive, they must be mixed. The largest axle torques were measured for deep subsoiling. Time portions of such heavy operations are important (Fig. 5.137 below).

Regarding the next step, forecasts of the S-N curves, converted in torque scale, contain the largest uncertainties [5.106].

Table 5.29: Integrated system of calculation and testing transmission gear wheels, based on cumulative damage analysis. *The largest uncertainty is not the damage calculation* (Fig. 5.132) but is the forecast of the S-N curves scaled in torque and modelled for *mounted* gear wheels


Input load spectra in sum frequencies log-log, probability 90-95%
Torques at low speeds
Torques at heavy duty speeds
log torque
log Σ time
Torques at transport speeds
Typical speed distribution
% of time
nominal speed
Technical specifications for transmission and all gear sets.
Required life, hours, & probability (e.g. B10)
Specifications to be modified
Load spectrum with life for each transm. speed including the constant gear sets
Total load spectrum for each gear set
Calculation of all torque endurance limits (from stress limits, e.g. B10) for basic criteria of all gear sets considering all local operational conditions
Cumulative damage algorithm
Cumulative damage evaluation for each gear set regarding the four basic criteria
"S-N" curve (load!)
Torque endurance limit
Haibach leg
Torque load T
Cycles N = contacts
Lab test design verification, typical strategy:
Step 1: Verification of S-N curves in load scale
Step 2: Verification of life applying load spectra by accelerated tests, deleting cycles of low loads which do not damage etc.
©Renius
Evaluation
Not optimal
Release
feedback
feedback
Feedback of test results improving calculations, mainly for "S-N" curves

5.8.4 Standard load spectra

The gearbox input load spectrum of **Fig. 5.133** summarizes all operations with a probability of 90–95%. *Example* for reading the diagram: 10% of total service time represents all torque loads above or equal to rated torque (not maximum torque). The average is equal to 0.545% of rated. With an average of about 75% rated speed, this equals about 41% rated input transmission power.

The author thinks that this spectrum is still suitable worldwide with two amendments:

1. Load increase in the case of *automatic power control* [5.114], see the dotted line.
2. Load increase for *higher torque backups*: Every % >15 % should be respected by a 0.5 % increase in load.

Load spectra for heavy-duty operations are typically round bodied [5.21, 5.28 and 5.115]. **Fig. 5.134** shows a typical load spectrum for heavy plowing [5.115], inserted in the above diagram with a 30% time fraction. Average torque was in this case 76% of rated. This special spectrum copes well with the total spectrum, which has still reserves to incorporate other individual load cases.

Cumulative damage evaluations require class frequency load spectra (histograms), which is demonstrated by converting Fig. 5.133 into **5.135**. Class width should not be too large, at least as shown.

If we apply this histogram for bearings using equation (5.19) we get T_{eq} = 72% of rated input torque for ball bearings and 74% for roller bearings.

Slightly higher values result if the amendments are applied.

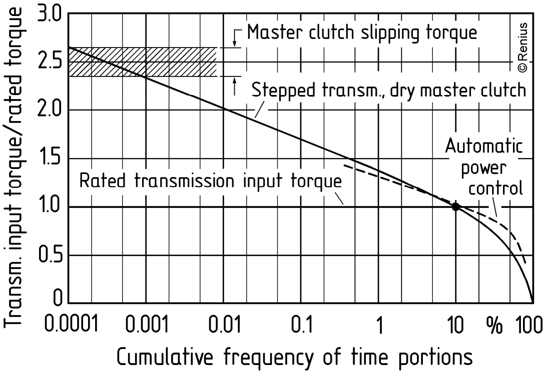


Fig. 5.133: Load spectrum "Renius" [5.21] for the input shaft of stepped transmissions, max. 15% torque backup. Dotted line for automatic full power control [5.114]

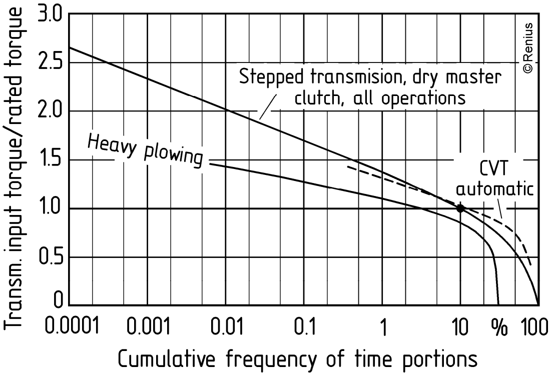


Fig. 5.134: Comparison of a measured spectrum for 30% heavy plowing with the total load spectrum [5.115]

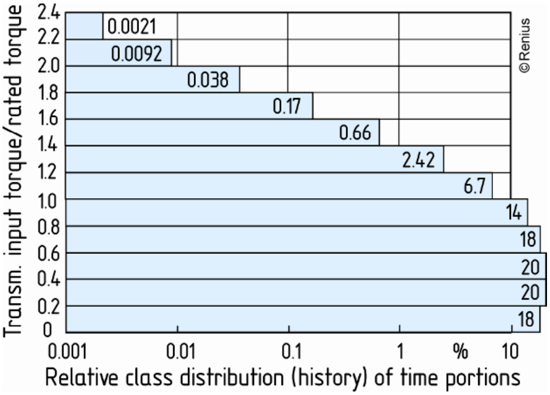


Fig. 5.135: Load spectrum of Fig. 5.133 [5.21] in class frequency distribution (also called "histogram")

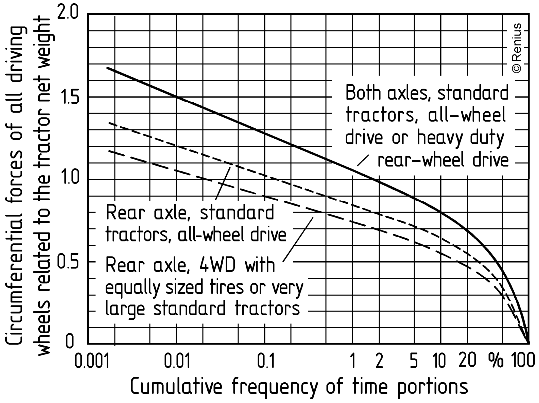


Fig. 5.136: Axle-torque load spectra for tractors, originally defined for heavy duty operations of small standard tractors with rear-wheel drives [5.21], probability about 95%

In 1976, a first standardized axle torque load spectrum was published in [5.21], **Fig. 5.136**. At that time, it addressed tractors with rear-wheel drives and 30–110 kW rated engine power (continuous line). The standardization of the ground drive torques with the nominal tractor net weight seemed to be slightly simple but proved over decades to be the best method for generalizing the spectrum.

The use of circumferential forces eliminates the tire radii. The spectrum is based on a broad publication review including measurements from KHD (Deutz). Furthermore, tests on this basis were compared with the long-term customer satisfaction of mass-produced transmissions of Deutz and several other manufacturers.

Meanwhile, typical working speeds have increased, resulting in decreased torque levels. Four-wheel drives have become the standard version for upper technology levels. The spectrum, initially defined for rear-wheel drive, is now useful for modern 4WD tractors as well; however, is still also valid for rear-wheel drive tractors which operate under conditions as they were typical in mid-Europe in 1960–75.

Deep subsoiling on a dry surface creates very good traction conditions, in particular if the resulting implement force crosses the surface between the axles, **Fig. 5.137**.

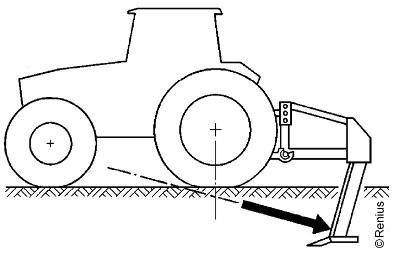


Fig. 5.137: Subsoiling with high depth on dry soil enables very high traction forces and torques; in particular, if the implement force is crossing the surface between the axles and if the tillage wings or plates are PTO-driven

The driver may be happy to have high pull forces at very low speeds.

However, this generates extremely high torque loads, mainly at the final drive, which are, for example, only covered for 200–300 hours by the standard load spectrum.

Long-time operations of this kind can require heavy-duty axles or electronic power limitations.

Rubber belt drives can develop higher traction coefficients than 4WD tractors. Measured axle torque load spectra of a Challenger CH45 were reported in [5.116]. The total operational time fraction of deep subsoiling was limited. Under this condition, the total axle load spectrum was very similar to that of Fig. 5.136. This was surprising but could be explained by high working speeds.

However, measured gearbox input load spectra of the Challenger [5.116] had a higher load level than that of Fig. 5.133. The reason is the high torque back-up of 42%, confirming the amendments made on the previous page.

5.9 Transmission efficiencies

Tractor prices are widely based on rated engine power. The higher the transmission efficiency, the better the use of the investment and the lower the CO2 emissions.

High efficiencies have, however, further advantages. They generate lower cooler costs and less additional energy for cooling fans. Tractor transmission efficiencies must be properly defined regarding input and output locations, which is sometimes missed.

The author recommends defining tractor transmission efficiencies by dividing axle output power by transmission input power.

The total losses are usually regarded to be the sum of two groups [5.103, 5.117, 5.118]:

- *load-dependent* losses
- *load-independent* losses

Both can be kept low by many design details, **Table 5.30.** Influences of input speed, load, oil temperature, oil level, and tractor speed are presented in [5.96, 5.118]. Reduced teeth heights can reduce the sliding losses considerably [5.119], but can need high numbers of teeth.

Load-dependent losses of gears are low, **Table 5.31**. They can be calculated well based on friction mechanics [5.103, 5.120].

Load-independent losses are difficult to calculate. It is therefore recommended to measure these losses for zero load level.

As an average, a *total loss* of 1.5% per spur gear contact, 1.7% per helical gear contact, and 2.5% per bevel gear contact can be used as a rule of thumb.

A good tractor transmission should not have more than six contacts (input to axle outputs) for speeds of the main working range and lower for higher speeds. Transmission efficiencies can be plotted similarly to engine efficiency maps, as presented by *Reiter*, **Fig. 5.138** [5.96]. The trends are confirmed for small tractors by [5.121].

Table 5.30: Design fundamentals and rules for minimized transmission losses

- Transmission designed for low oil level
- Optimal oil viscosity specified
- Pre-heating in cold regions possible
- Forced oil lubrication, ideal: flow on demand
- Low rotational speeds of elements
- Low speed differences at sliding surfaces
- Low number of gear wheel engagements
- Reduced teeth heights (limits: contact ratio)
- Enclosures for elements touching oil level
- Properly adjusted tapered roller bearings
- Seals with low friction
- Low brake idling losses, sufficient clearance between plates of disengaged wet brakes
- Possibililty of disconnecting unused transmission elements, for example 4WD and PTOs
- Low surface roughness of gear wheels
- Planetaries instead of countershaft drives
- Installation of an efficiency monitoring system

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Table 5.31: Load dependent losses of spur gears (one engagement), derived from [5.96]. Values for helical gears about 0.2% lower

Rule of thumb
1.4 ..... 1.0 ..... 0.7%
← ratio
← specific load
← surface roughness
circumfer. speed →
oil viscosity →

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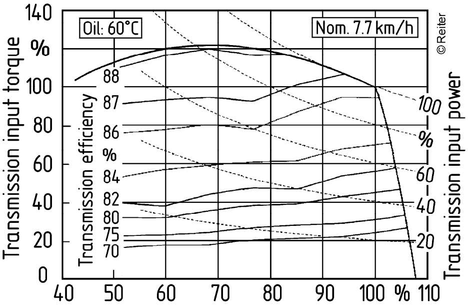

Fig. 5.138: Measured efficiencies for a stepped tractor transmission of the tractor Fendt Farmer 310 LSA [5.96] (100% power = 63 kW). Gearbox oil SAE 80, final drive oil (spur gears) SAE 90, fluid coupling dismounted, dry disc brakes. Transmission was used for tractors up to 85 kW

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German **Yearbook of Agricultural Engineering,** founded 1988 by *H.J. Matthies*, TU Braunschweig. Editors (first person always full professor at TU Braunschweig):

- 1988–2004 *H.J. Matthies* and *F. Meier*
- 2005–2009 *H.H. Harms* and *F. Meier*
- 2010 *H.H. Harms*, *F. Meier*, and *R. Metzner*
- 2011 *H.H. Harms* and *R. Metzner*
- 2012 until now (2019) *L. Frerichs*

This yearbook contains annual reviews on tractors and its components. The chapters on tractors and those on engines and transmissions were presented uninterrupted by the author, often in cooperation with co-authors or as co-author, issues 1991–2010 bilingual German-English.

Issue 1994 is formerly not existing as the volume counting method was modified (no break).

Issues are charge-free available since 2012 by open access thanks to Prof. Frerichs, supported by the whole German ag eng community.

6 Tractor workplace

6.1 Introduction and workload on operators

The scientific investigation of human-machine relations is still a young discipline [6.1–6.5]. Its main subjects are:

- *Health*
- *Safety*
- *Comfort*

The first steps towards cabs were optional compartments with fluttering tarpaulin elements protecting the driver against wind and rain. They became slightly more popular in combination with safety frames (compulsory in several countries, in Germany for newly registered tractors from 1970).

Another add-on was the optional equipment of simple closed cabs which were mounted directly on the tractor chassis.

However, they could not reduce the noise level at the driver's ear, which remained at about 90–100 dB(A) under full load. This level damages the ear; thus, is not tolerable from a human standpoint.

The breakthrough of modern cabs took place in the developed countries during the early 1970s although cabs are one factor of increased initial costs of tractors, **Fig. 6.1**.

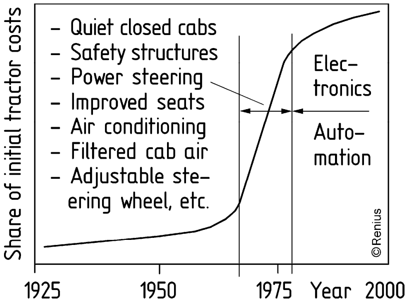

Fig. 6.1: Progress in human relations and specific initial costs for driver's health, safety, and comfort for upper tech levels

John Deere presented its first noise isolated cab, Sound-Guard®, in 1972 offering a full load noise level at the driver's ear of only 82.5 dB(A) [6.6] and, in addition, an integrated safety frame for protecting the driver when overturning.

However, even today such cabs or safety frames are often not applied in developing countries [6.7]. Typical comments are:

"will not pay",
"not competitive in costs",
"more parts – more problems", etc.

Therefore health and safety authorities have initiated or even forced first steps in several countries. This is the case, for example, in Western Europe for noise levels which by law must not exceed 90 dB(A) at the driver's ear in a tractor under full load. Also safety frames must be installed by law. There were hot discussions on the additional costs, but this was finally a well done political act.

After a certain period of time, farmers recognized the benefits of improved health, safety, and comfort; the noise isolated safety cab became even wanted.

Along with this trend, tractor companies began to compete in advertising low noise values reducing them to now only about 69–76 dB(A) at the driver's ear for technology levels IV and V with typical values of 80–85 dB(A) for low-cost cabs.

A remaining problem is the equipment of small tractors with good cabs. Fundamentals of ergonomics do not correlate with the tractor power or size as driver dimensions are always the same.

This hinders cab introductions for small tractors as initial cab costs in relation to total tractor costs are up to almost 20 %, while this value declines with increasing power level down to only about 10% for very big machines.

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https://doi.org/10.1007/978-3-030-32804-7_6

Workloads on the operator are usually classified into four sections, **Table 6.1**, each with characteristic criteria. The total workload is a result of all these particular factors and is balanced by the operator's potential and willingness to work.

Physical and mental stress creates "human fatigue" which reduces human performance.

The risks of accidents are also increasing; however, consequences can be diminished by "design for safety" (see, for example, safety frames protecting the driver in case of overturning, chapter 6.6.4).

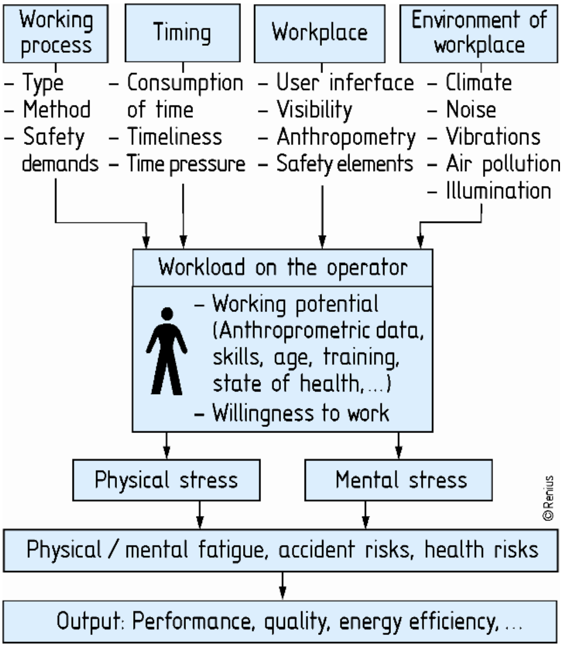
Stress which exceeds individual stress limits for longer periods of time can damage health. Typical examples are intervertebral disc problems due to vibration overloads, or defective hearing as a consequence of high noise levels at the driver's ear. The second aspect is very important as this type of occupational injury cannot be healed.

The goals for the driver can be identified by general criteria such as productivity, quality of work output, energy efficiency, machinery wear, maintenance costs, comfort, health, safety, driver downtime and others. Some of these criteria are difficult to measure; however, the operator can set targets for particular conditions.

Modern digital systems are able to optimize the total process of tractor and implement.

The two most important strategies are
- *use maximum engine power*
- *optimize energy efficiency* (mainly for partial load)

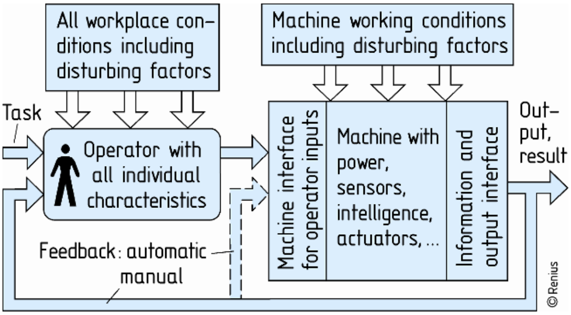
Table 6.1: Operator performance (from [6.3], modified)



The function of an operator-machine system can be understood as a *closed loop control* circuit, **Fig. 6.2**. An important element is the feedback of output results to the operator, enabling corrections.

Further reductions of the workload are possible if the feedback and correction are automated, bypassing the driver who is now only setting the target; see, for example, the automatic three-point hitch control or automatic guidance systems.

Table 6.2: Closed loop control of an operator-machine system (e. g. tractor-implement system) [6.8]


All workplace conditions including disturbing factors
Machine working conditions including disturbing factors
Task
Operator with all individual characteristics
Machine interface for operator inputs
Machine with power, sensors, intelligence, actuators, ...
Information and output interface
Out-put, result
Feedback: automatic
manual
©Renius

6.2 Technical aids reducing operator loads

Workloads and stress on the operator can be reduced considerably by technical aids, supported by standards, **Table 6.3**.

Important international standards were developed with recommendations and test methods, some even dealing as a basis for legal acts. ISO standards lead to acceptable workplace dimensions which must not increase costs; hence, are recommended even for lower technology levels.

Table 6.3: Driver workloads, technical aids, and related international ISO standards. Early proposals of this basic classification were published by *W. Batel* [6.9]

Workload categories		Technical aids for the operator and related international standards (ISO)
Working process	Set tasks	- Instrumentation, on-board computer (ISO 11783), touch screen
	Compare tasks with output	- Field of vision with minimum obstracles (ISO 5721) - Connecting 12V sensors and instrumentation (ISO 11786) - Application of on-board computer, GPS-based (ISO 11783) - Use of modern displays, standardized symbols (ISO 3767)
	Correct process	- Ergonomic proportions between steering wheel, seat, levers, pedals, and seat adjustments (ISO 4253). Definitions of SRP and SIP see ISO 5353 - Low actuating forces, ergonomic directions of lever and pedal movements, coding by colours (ISO 15077) - Standardized symbols (ISO 3767) - Shift aids: synchronizers, power shifts, automatic shifts - Cab aperture for external equipment controls (ISO 8935) - Safe cab access, internal cab clearances, hand clearances at movable control elements (ISO 4252)
Environment of workplace	Climate	- Thermal comfort (ISO 14269-2): Air conditioning balancing solar heating. Adequate heating capacity, low air speeds. Evaluation by tests (ISO 14269-2)
	Noise	- Engine and cab suspended on silent blocks - Design rules of ISO 11688-1 and -2 to be applied - Noise measurement inside vehicles (ISO 5128) - Low noise level at driver's ear, measurement (ISO 5131) - Low noise outside, measurement (ISO 362)
	Dust and chemicals	- Pressurized air inside cab (ISO 14269-5) - Filtered air input (ISO 14269-4, ISO 16890) - Special filter types for chemicals (EN 15695)
	Vibrations	- Passive reduction of human accelerations by springs and dampers for seats, cabs, axles, mounted implements, etc. - Active driver position control by hydraulics or electrics - Evaluation of human body vibration by ISO 2631, seat evaluation by ISO 5007 (lab), vehicle evaluation on smooth track ISO 5008, recording of road surfaces by ISO 8608
	Visibility	- Working lamps for night, good illumination in the cab - Cameras for improved visibility and safety (ISO 29112-1)

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Hand and pedal forces depend mainly on their *location* and *direction of movement*, **Fig. 6.2**, **6.3**, and **6.4**. Maximum values are not suitable for continuing use. Hence, ISO 15077 falls below them, **Table 6.4**.

High comfort can require even slightly lower forces, mainly for levers.

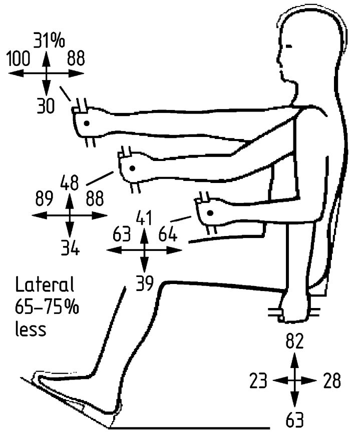


Fig. 6.2: Relation of hand forces in % [6.5].
Scale: Average max. male: 100% = 614 N
Average max. female: 100% = 440 N

Table 6.4: Maximum forces for control elements according to ISO 15077

Type of control	Max. force, N
Lever fore/aft	230
Lever lateral	100
Lever vertical upwards	400
Pedal-leg/foot operation	450
Pedal-ankle rotation	90
Finger/wrist operation	20

Steering and braking forces not included, are specified in ISO 10998 and ISO 5697

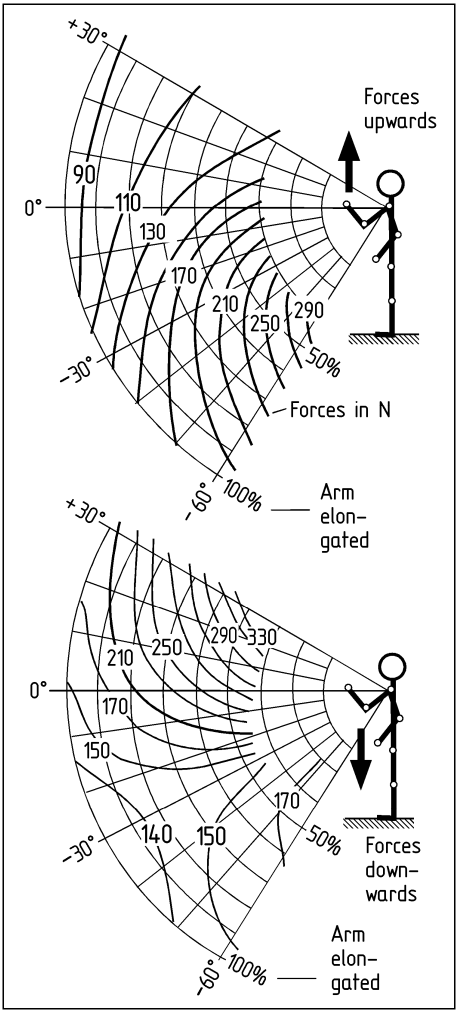


Fig. 6.3: Maximum vertical hand forces of a male person in Newton according to *Rohmert* [6.9]

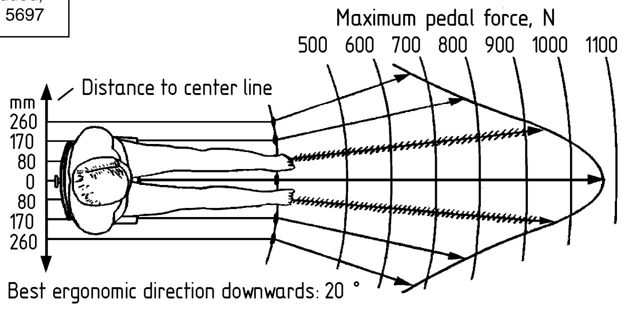


Fig. 6.4 (right)**:** Maximum pedal forces of a male person in a seat with a stable backrest.
Adapted from *Dupuis* [6.10]

6.3 Operator's seating accommodation, access, and exit

Design of workplaces should be based at least on relevant ISO standards.

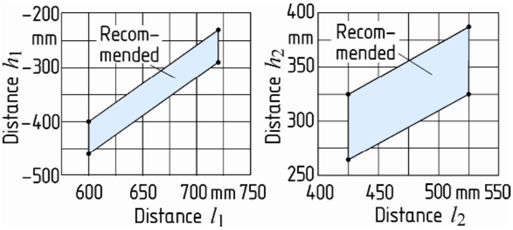


Fig. 6.6: Recommended relations of l_1 to h_1 and l_2 to h_2 according to ISO 4253. Seat Index Point SIP is defined in ISO 5353 by a loading apparatus. SIP replaces the Seat Reference Point SRP which was standardized in ISO 3462 (withdrawn)

Basic dimensions of ISO 4253 are summarized in **Fig. 6.5** and adjustments for different human body sizes in **Fig. 6.6** and **Table 6.5**.

Safety and comfort for the driver are also influenced by access to and exit from the cab and its interior clearances. Minimum dimensions are shown in **Fig. 6.7**, based on ISO 4252.

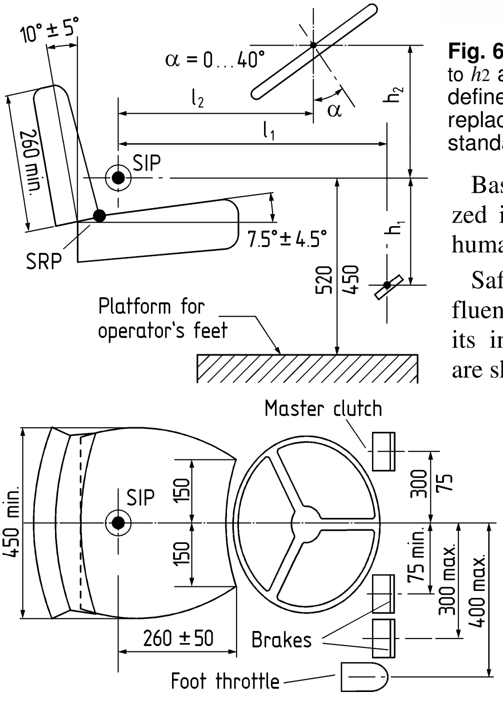


Fig. 6.5: Working place dimensions (ISO 4253)

Table 6.5: Seat adjustment as recommended in ISO 4253

Seat adjustment	Range Min. mm	 Opt. mm
Longitudinal adjustment forward-rearward from the mid position	± 75	± 100
Vertical adjustment up-down from the mid position	± 30	± 50
Instructional seat see ISO 23205		

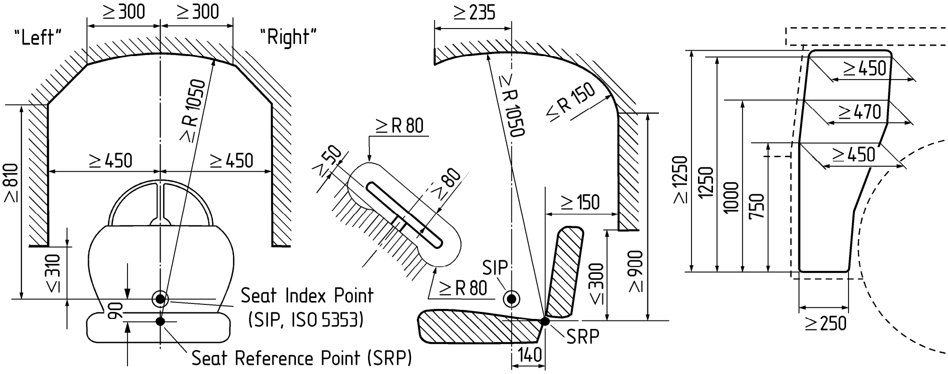


Fig. 6.7: Cab clearance and access-exit dimensions as recommended in ISO 4252

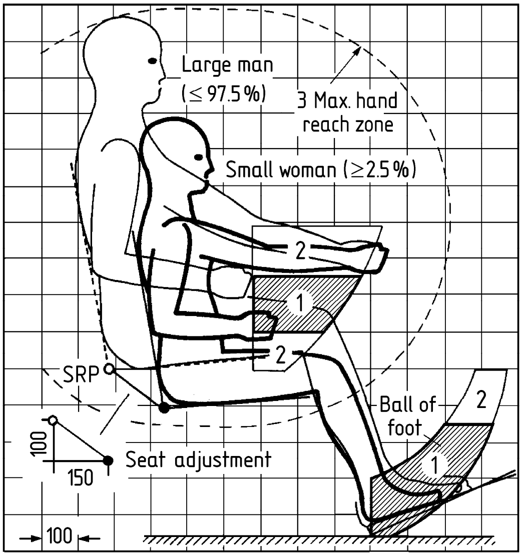


Fig. 6.8: Hand and foot control positions, side view. Extract from [6.5] for US adults.

Comfort rating: 1 – excellent, 2 – good, 3 – fair.

Seat adjustment for this (very wide) range of human measurements is still within the values as recommended in ISO 4253 (Table 6.4)

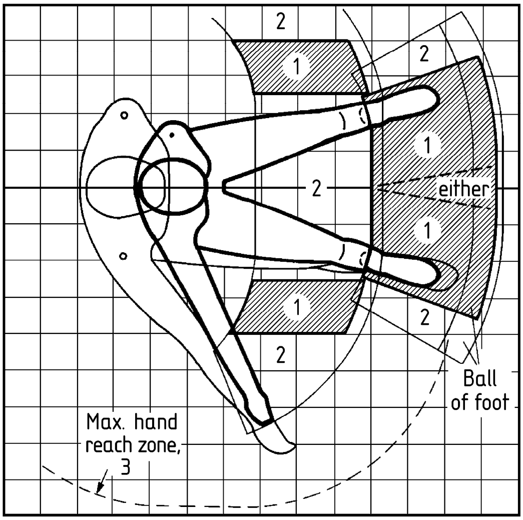


Fig. 6.9: Hand and foot control positions, top view. Extract from [6.5] for US adults.

Comfort rating: 1 – excellent, 2 – good, 3 – fair

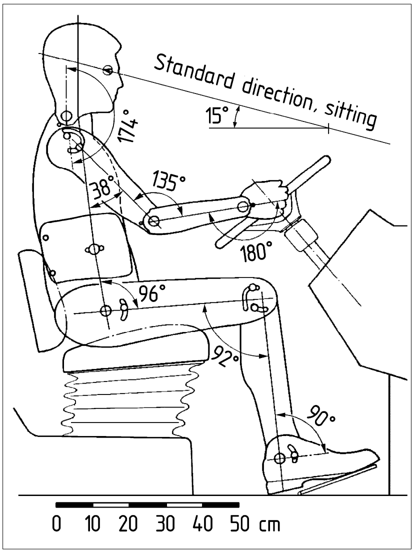


Fig. 6.10: Recommended angles between human body limbs, based on the body model of the German standard DIN 33 408

Measurements of the human body as well as angles of comfort were presented 1980 by *Purcell* [6.5] and are still useful.

Basic human dimensions are standardized in ISO 7250.

The recommended hand and pedal zones of the *Purcell* paper are summarized in **Fig. 6.8** and **6.9**.

Applying them to tractor cabs, hand zone 1 on the left side is usually in conflict with the main cab access and exit zone. This resulted in a general priority for hand zone 1 on the right. Comfort angles of human body parts are standardized in DIN 33 408. Recommended values for upright sitting are plotted in **Fig. 6.10**. This drawing also shows, that the normal *sight line of an upright sitting person has an inclination of about 15°*.

This is important for the forward field of view and the shape of the engine bonnet.

6.4 Ride dynamics

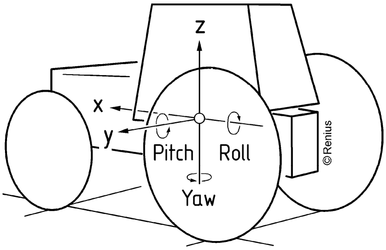


Fig. 6.11: Tractor axis system for defining vehicle dynamics (ISO 2631 and 8855).
Most important: z-movement, pitch, and roll

Ride dynamics can be evaluated by ISO 5008, depending mainly on

- *Configuration* of tractor and implement
- *Driving surface* (profiles in ISO 5008)
- *Tractor speed*
- *Technical aids* to reduce vibration

There are six basic movements, **Fig. 6.11**.

The inclusion of mounted implements is important for driving on the road, **Fig. 6.12**.

Inertia of mounted implements influences ride dynamics (and dynamic axle loads) considerably. Top link damper or integrated active damping systems are used.

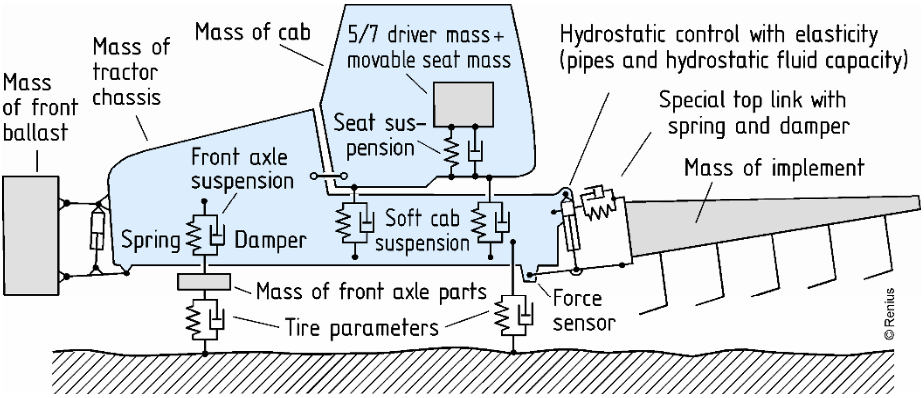
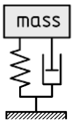
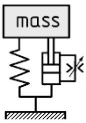
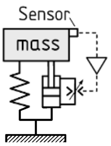
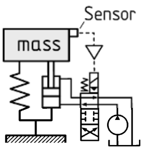


Fig. 6.12: Simplified (two-dimensional) dynamic model of a tractor with hydro-pneumatic front axle suspension, soft cab suspension, and passive implement damping. From [6.12], modified

Table 6.6: Reducing driver vibration (after *Hauck*)

Passive system	Adaptive system	Semi-active system	Fully active system
 	 	 	
No control	Open loop control	Closed loop control	Closed loop control
No control enegy		Low control energy	High control energy

Riding comfort is mainly influenced by three modes of movement with the following *typical natural frequencies* of common two-axle tractors:

- ***Moving up and down*** (**2.5–3.5 Hz)**
- ***Pitching*** "tractor only" **(1.8–2.5 Hz)**
- ***Pitching*** w. mounted impl. **(0.9–1.5 Hz)**
- ***Roll*** (around x-axis, **1–2 Hz)**

Technical aids for reducing the accelerations for the driver can be classified into four groups, **Table 6.6**.

The technical effort increases from the left to the right; however, also do initial costs and energy consumption.

Mechanics of a simple *Voigt-Kelvin* element are demonstrated in **Fig. 6.13**.

If there is no damping, the natural or resonance frequency ω_0 in rad/s is

$$\omega_0 = \sqrt{\frac{c}{m}} \qquad (6.1)$$

where

c = spring rate [N/m], m=movable mass [kg].

The frequency f in Hz [cycles/second] is

$$f = \omega/2\pi \qquad (6.2)$$

If the system is dampened, the natural frequency is slightly lower than ω_0:

$$\omega_0' = \sqrt{\frac{c}{m} - \left(\frac{k}{2m}\right)} \qquad (6.3)$$

where k = damping coefficient, which is defined by the damping force F_D.

$$F_D = k\,[\text{Ns/m}] \cdot \text{damping speed [m/s]} \qquad (6.4)$$

If the mass m in Fig. 6.13 is 5/7 *driver mass plus movable seat mass*, the vertical driver amplitude can only be decreased if its natural frequency is at least 29% lower than the input amplitude of the cab bottom.

Estimation of natural tractor frequencies. We need at least the tractor mass distribution and undercarriage spring rates.

Spring rates (stiffness rates) of tires are influenced by the following factors:

- Tire size and tire design, bias or radials
- Inflation pressure
- Tire load or deflection
- Rolling speed
- Tire age

Stiffness rates versus displacements are presented in **Fig. 6.14** and **6.15** [6.13].

Example: A 110 kW standard tractor with a mass of 7,000 kg has a total vertical linear tire stiffness of 4 x 350 = 1,400 kN/m (radials, 1.1 bar). Equations (6.1) and (6.2) deliver a natural vertical tractor frequency of **2.25 Hz**. Resonance damping by the seat requires a 29% lower seat-driver resonance frequency (Fig. 6.13), means < **1.60 Hz**.

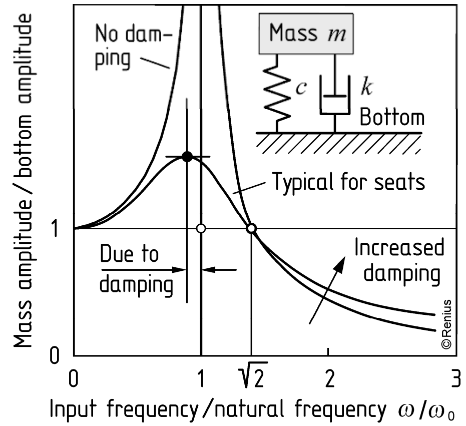


Fig. 6.13: Transfer function of a Voigt-Kelvin element with mass m, spring c and damping coefficient k

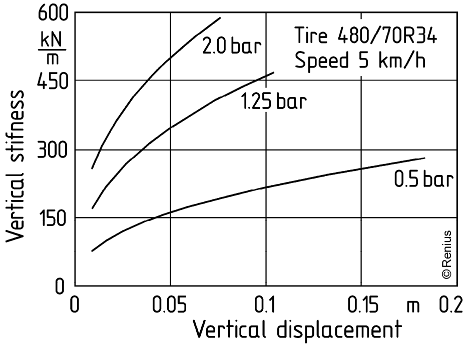


Fig. 6.14: Vertical stiffness of a popular radial tire, size 480/70R34 [6.13]

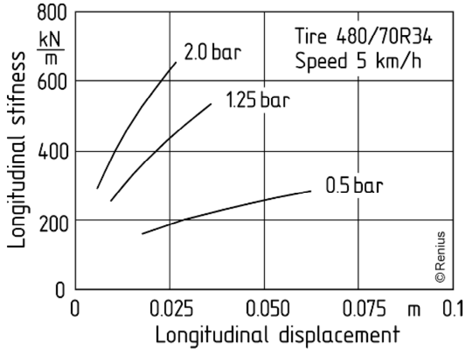


Fig. 6.15: Longitudinal stiffness of a popular radial tire, size 480/70R34 [6.13]

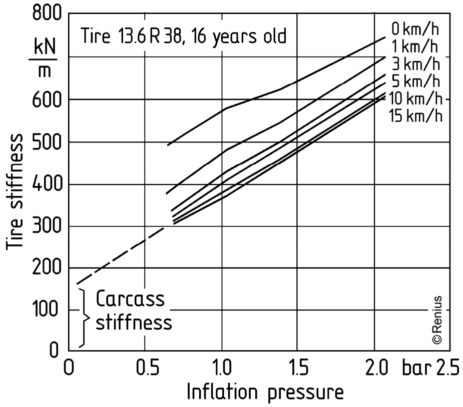


Fig. 6.16: Measured vertical tire stiffness versus inflation pressure and vehicle speed [6.14]

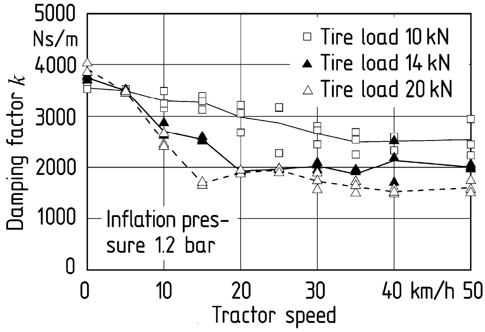


Fig. 6.17: Damping coefficient k of tractor tire 520/70 R38 versus tractor speed [6.15]

Stiffness rates versus inflation pressure were measured at Silsoe Research Institute (UK), **Fig. 6.16** [6.14]. Stiffness increases almost linearly with pressure but decreases up to speeds of about 10 km/h. These values are slightly above others due to the small tire width and their high age. If they are extended to the left they deliver the carcass stiffness. 150 kN/m for "rolling", listed in [6.14], is a high value in this case.

Tire models including damping. General trends of dynamic behaviour of tractor tires versus speed and inflation pressure were already summarized in chapter 3 based on publications from TU Berlin (under *Göhlich* and successor *Meyer*).

Tire research was also carried out at the University of Hohenheim under *Kutzbach* and his successor *Böttinger.*

Fig. 6.17 shows typical damping factors k of a radial tire versus speed for different tire loads at constant inflation pressure. The higher the load and the speed, the lower the damping.

The Hohenheim Tire Model, **Fig. 6.18**, was proved to deliver realistic three-dimensional tire characteristics, if adequate values are introduced for the spring rates and damping factors [6.16, 6.17].

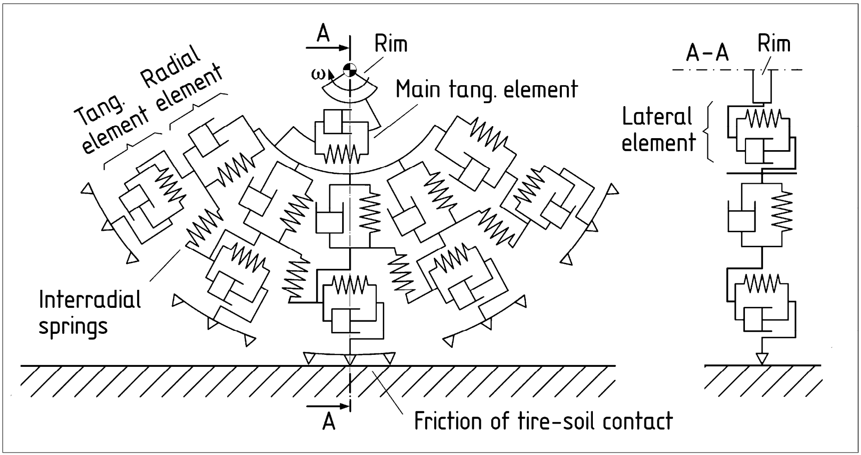


Fig. 6.18: Structure of the "Hohenheim Tire Model (HTM)" as of 2018 [6.16, 6.17]

Reducing pitching. Pitching is important for driving tractors on-road with heavy rear-mounted implements, mainly at high speeds and on bad roads, **Fig. 6.19**.

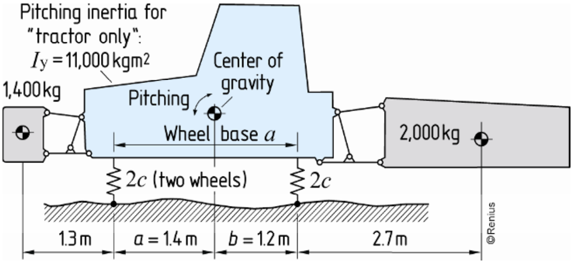


Fig. 6.19: Simplified model of a 110 kW tractor, 7,000 kg net weight, with front ballast and rear-mounted implement

If we assume the center of gravity as indicated in Fig. 6.19, the calculated additional moment of inertia for the front weight is

$$I_{1400} = 1{,}400\,\text{kg} \cdot 2.7^2\,\text{m}^2$$
$$= 10{,}206\,\text{kgm}^2$$

and the appropriate value for the rear mounted implement:

$$I_{2000} = 2{,}000\,\text{kg} \cdot 3.9^2\,\text{m}^2 = 30{,}420\,\text{kgm}^2$$

Total inertia I is the sum of the three mentioned values, stiffness c stands for one tire. ω_0 (without damping) [6.4] results in:

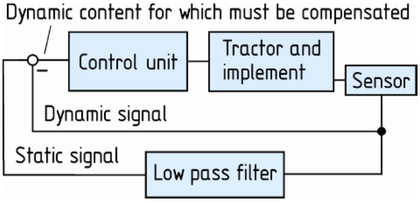
$$\omega_0 = \sqrt{\frac{2c\cdot(a^2+b^2)}{I}} \qquad (6.4)$$

Example: $a = 1.4$ m, $b = 1.2$ m, $c = 350{,}000$ N/m (equal tire stiffness, 1.0–1.2 bar inflation) and total inertia $\boldsymbol{I = 51{,}626}$ kgm^2.

$\omega_0 = 6.79$ rad/s and $f = \omega_0 / 2\pi = \mathbf{1.08\,Hz}$.

Pitching can create problems of comfort, axle dynamics, and steering safety, which has initiated many research projects [6.18–6.27]. Top links with *integrated damper* (Fig. 6.12) were found to be an effective *passive* solution [6.18]. If the tractor has draft force sensors at the three-point hitch (Fig. 6.12), their signals can be used for *active damping*, **Table 6.7** and **Fig. 6.20**.

Table 6.7: Active pitch damping by closed loop control of lower link forces [6.23]



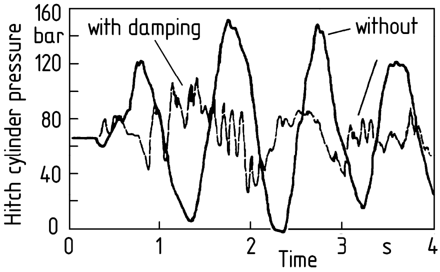


Fig. 6.20: Active implement damping, indicated by the hydr. hitch cylinder pressure [6.23, 6.25]

Such a system was first presented in 1989 by *H. Hesse* (Bosch) and *H. Göhlich* (TU Berlin) [6.22 - 6.25], first applied in 1989 to a Fendt tractor. It uses the digital electronic rear hitch draft control, is very simple, and soon became popular. Fig. 6.19 confirms the damping and the low pitching frequency; in this case about **1.1 Hz**.

Driver limits of vertical body acceleration [6.28] are plotted in ISO 2631, **Fig. 6.21**.

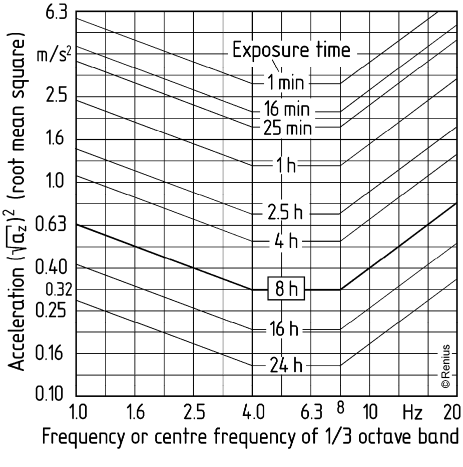


Fig. 6.21 (right)**:** Human limits of vertical body acceleration, based on ISO 2631-1

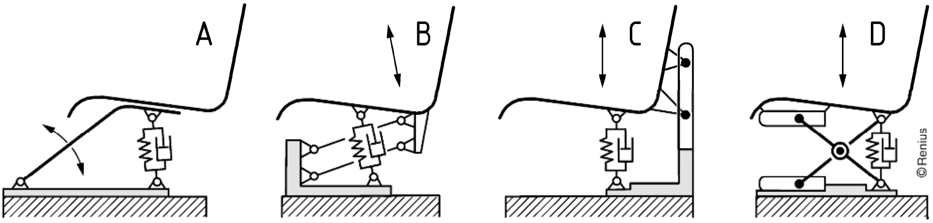


Fig. 6.22: Four typical design principles of suspended tractor seats with springs and dampers. Design A is only used for lowest technology levels because of the seat rotation and bulkiness

Table 6.8: Typical specifications of tractor seats (simplified, kindly supported by GRAMMER AG)

Technology Level	Basic	High	Very High
Vertical suspension			
Travel, max., mm	80	100	120
Nom. nat. frequency	1.8 Hz	1.4 Hz	1.1 Hz
Spring system	mech.	pneumatic	
Driver weight, kg		50-150 kg	
Weight adjustment	manual	autom.	electron.
Damping	constant	adjustable	
Damping adjustm.	-	mech.	autom.
Longitudinal susp.	-	mech.	
Lateral suspension	-	mech.	
Height adjustm., mm	60	80	
For/aft adjustm., mm	150	210	
Positioning	-	automatic	
Energy supply	-	12V (compressor)	
Seat swivel (z axis)	-	20°	40°
Cushion width, mm	450	500	550
Seat climate control	-	passive	active
Armrest	-	std.	std.*)
Seat belt (ISO 3776-1)	optional		std.
Operator presence control (OPS)	-	opt.	std.
*) with interface for multifunctional armrest			

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Tractor seats are offered in a wide variety of designs and functions, **Table 6.8**.

Comfort level and initial costs increase considerably from left to right. Natural frequencies vary from about 1.8 to about 1.1 Hz (left to right), based on

movable seat mass + 5/7 driver mass

Low spring rates and optimized damping rates deliver low natural frequencies (equation 6.3).

Unacceptably high vertical amplitudes can also be reduced by non-linear, adaptable spring rates and automatic damping control [6.28].

High-tech seats are often equipped with multifunctional armrests, moving together with the driver, offering tractor and implement control by fingertip, often in combination with touch screen monitors.

Highly sophisticated seats improve tractor ride comfort for all tractor movements (x, y, and z directions); however, pitching is reduced mainly by implement damping as mentioned above.

A modern seat is shown in **Fig. 6.23**.

If the actual frequencies of the tractor motions are close to the natural frequencies of the driver-seat unit (Fig. 6.13), the seat delivers poor damping.

Therefore, semi-active or even fully active seat suspensions, front axle, and cab suspensions were developed for tractors.

Fig. 6.23: Seat GRAMMER MSG 95A/731, representing level "high" of Table 6.8.
Photo courtesy GRAMMER AG

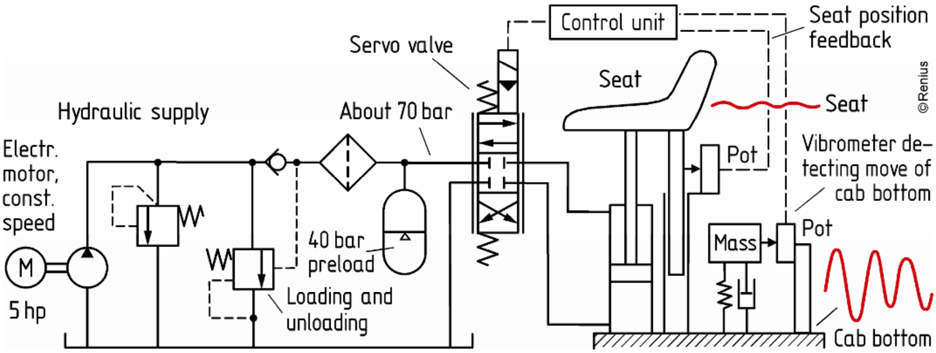

Fig. 6.24: Active seat suspension system, prototype USA 1968 [6.32]. Symbols ISO 1219

The performance of tractor seats can be improved by semi-active control or fully active systems, as mentioned above.

Simple *damping control* can be realized by viscosity control at the damper piston with rheological fluids [6.29, 6.30].

Even better results are achieved by fully active systems; for example, as had been demonstrated by *Stikeleather* as early as 1968 [6.31]. He stated in [6.32] that a successful control was achieved with the system of **Fig. 6.24**; however, with the weak point of high energy consumption of the hydraulics.

In 2002, John Deere presented the Active Seat™, the first commercial active seat suspension system [6.33], **Fig. 6.25** (left), obtaining its energy from the on-board hydraulics for pump supply pressures between load sensing standby (about 35 bar) and maximum pressure (for tractor systems usually about 200 bar).

This was a large step forward, but energy consumption was still high with an average of, for example, about 1 kW.

This disadvantage was reduced by the second generation, the eActive Seat™, introducing a highly dynamic electric drive, **Fig. 6.25** (right), consuming much less energy. Its output shaft drives the scissors via a planetary counter-rotating (with speed reduction), correcting the vertical driver position by minimizing the (measured) vertical acceleration [6.34].

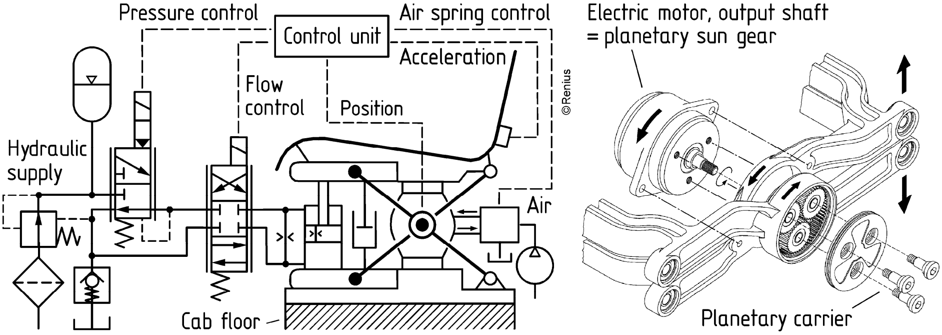

Fig. 6.25: John Deere Active Seat™. At left the first generation with hydraulic energy supply as explained in [6.33] and in patent DE 601 21 688 T2 (Priority 5.9.2000).
At right the central unit of the improved generation eActive Seat™ as described in the further patent US 8,622,362,B2 (issued Jan. 7, 2014) [6.34]

6.5 Tractor noise fundamentals

Fundamentals of acoustics can be found in [6.35]. There are two important types of tractor noise levels:

- Noise level at the *driver's ear* (ISO 5131)
- Noise level *outside* (ISO 7216)

Noise levels at the driver's ear were investigated early on, e. g. by *Matthews,* et al. [6.2]. Practical levels on tractors show a broad spectrum, **Fig. 6.26** [6.36].

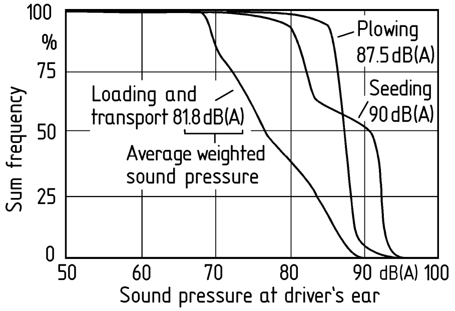

Fig. 6.26: Measured sound pressure collectives for three tractor operations. Tractor with an isolated, but simple low-cost cab [6.36]

As a simple rule of thumb, a sound pressure level of 85 dB(A) under full load should not be exceeded at the driver's ear during a day. For details see ISO 1999.

Sound pressures are expressed in effective root-mean-square values. The lowest effective sound pressure which can be detected by human ears is 20 µPa and was used to define the *noise level zero dB* [6.35, 6.37].

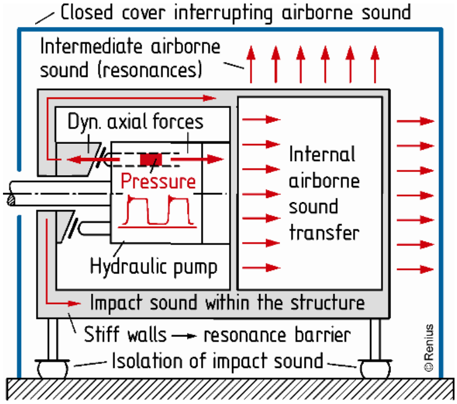
Closed cover interrupting airborne sound
Intermediate airborne sound (resonances)
Dyn. axial forces
Pressure
Internal airborne sound transfer
Hydraulic pump
Impact sound within the structure
Stiff walls → resonance barrier
Isolation of impact sound
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Fig. 6.27: Example for transportation of sound as generated by cyclic pump pressure

Sound pressure L_P is expressed by using a logarithmic scale (in dB):

$$L_P = 10\cdot\log\left(\frac{p_{eff}^2}{p_0^2}\right) = 20\cdot\log\left(\frac{p_{eff}}{p_0}\right) \qquad (6.5)$$

where

p_{eff} = effective pressure (root mean square)
p_0= reference value = 20 µPa

The "A" of the dB(A) respects the human reaction by filtering the sound pressures levels versus the related frequencies as standardized in IEC 61672-1 [6.38].

The sound emission level of a system can be expressed by the sound power level L_I (also "sound intensity level"), in dB:

$$L_I = 10\cdot\log\left(\frac{I}{I_0}\right) = L_P - 0.29\ \text{dB} \qquad (6.6)$$

where

I = actual sound power density in W/m^2
I_0 = reference value = 10^{-12} W/m^2

The total (absolute) sound emission of a system is the total sound power through its complete virtual envelopment. It can be measured independently of the distance between system and measuring points.

Sound emission can be controlled by five design principles, **Fig. 6.27**:

- Reducing sound generation at the source
- Damping impact sound of the structure
- Interrupting impact sound transport
- Damping airborne sound
- Interrupting airborne sound transport

Tractors in block chassis design without any application of these principles of noise reduction are usually not able to meet the 85dB(A) rule.

If a cab is rigidly mounted onto the tractor chassis, the noise level is not reduced, but usually increased. This typical mistake of many early designs can be prevented by putting the cab on silent blocks (Fig. 6.27).

How can **tractor noise emissions** be reduced? Cabs can protect the driver, but they are expensive, mainly for small tractors.

One possibility of reducing the noise without cabs is to abandon the block chassis design, and its impact sound of engine, transmission, and hydraulics which is emitted by the complete structure.

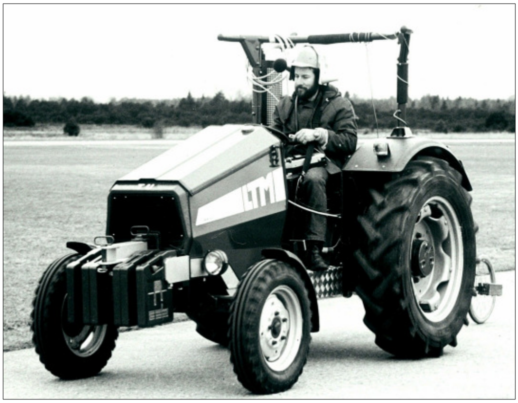

Fig. 6.28 (right)**:** The Munich Research Tractor, presented Feb. 25, 1988 at TU Munich, completely designed by *T. Kirste* (driver, Ph.D. thesis [6.40]) and *Karl-Th. Renius* (Head of Institute of Agricultural Machinery, TUM). The picture shows driving on the runway of the Königsdorf Airport south of Munich measuring noise levels with and without pulling loads. 84 dB(A) at the driver's ear could be achieved under full load at 7.5 km/h

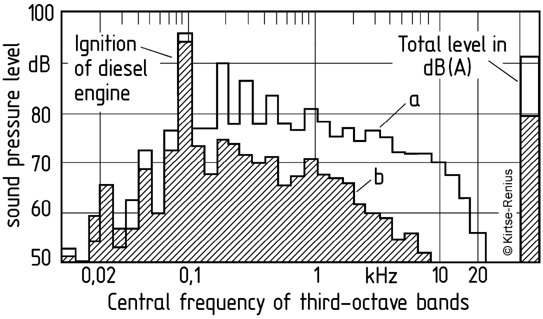

Fig. 6.29 (right)**:** Measured sound pressures of individual frequency bands (dB) and total sound pressure level in dB(A) at the driver's ear for the tractor of Fig. 6.28 (33 kW) at rated engine speed of 3000/min, stationary without load. No cab [6.40].

a Standard version, block concept
b Engine in silent blocks and encapsulated, only slightly open at the front (cooling air inlet) and at the bottom side (cooling air outlet), silenced transmission walls

Early research was carried out at TU Munich by *R. Bacher* [6.39] under *W. Söhne*.

Later on, the "Munich Research Tractor" was developed, **Fig. 6.28**, demonstrating a noise reduction of about 12 dB(A), **Fig. 6.29**, mainly by mounting the engine on silent blocks, using the hood as a noise damping capsule, and optimizing the transmission housing acoustics.

When the engine noise emission was considerably reduced, the transmission noise became dominant. Acoustic FEM analysis [6.40] demanded *higher stiffness for housing walls* to shift their natural frequencies considerably up above the excitation frequencies, see also the design of the ZF transaxle in **Fig. 6.30**.

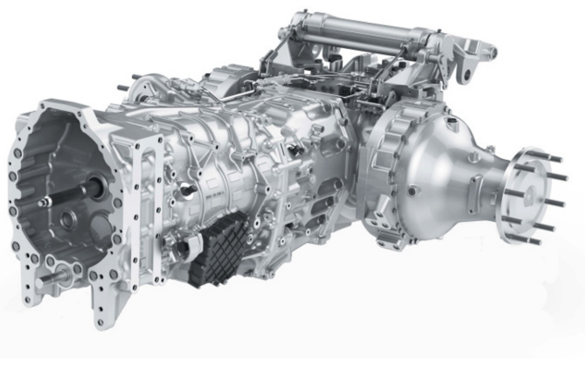

Fig. 6.30: Housing fins and stiff convexities on the ZF TMT 32 transaxle (2018). Courtesy ZF

6.6 Cab design

6.6.1 Design for low inside noise

Control of tractor noise emissions is mainly important for the driver as mentioned above. Noise emissions to the environment are less important, as the tractor is usually working in the field.

Table 6.9: Design rules for tractor noise reduction with particular reference to noise level at the driver's ear

- Engine in silent blocks (chassis frame)
- Engine encapsulated (holes for air)
- Optimized engine installation
- Low-noise engine cooling fan with speed control (power on demand)
- Optimized exhaust gas silencer
- Optimized position of muffler output
- Cab mounted on silent blocks
- No sound bridges from chassis to cab (if possible, complete drive-by-wire)
- Low-noise blower for cab air intake
- Low noise compressor for air conditioning
- Cab carefully airtight to keep noise putside
- Noise insulating mat between engine and cab
- Noise insulating mat under the cab bottom
- Noise insulating floor on the cab bottom (waterproof and easy to clean)
- Damping internal cab linings
- Dished, heavy glazing (not plane)
- Glazing in rubber settings
- Transmission in silent blocks (if possible)
- Torsional damper at transmission input
- Stiff transmission housings (fins)
- Helical gears with corrected profiles
- Hydraulic CVT units in silent blocks
- Special silencers within hydraulic pipes

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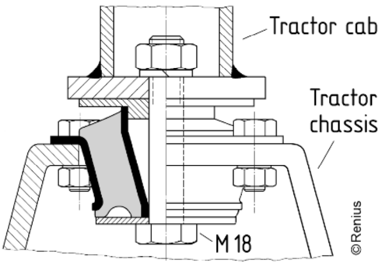

Fig. 6.31: Silent block for interruption of impact sound, typically applied for mounting of tractor cabs. Courtesy CNH (IH)

But there are traffic regulations in some countries (e. g., within the EU) with limits for bystander noise levels, when the tractor is passing by at a specified distance.

Design principles for meeting both requirements are listed in **Table 6.9**. The noise level at the driver's ear is in the foreground; it is influenced by acoustics of the whole tractor structure**.**

An e*ngine in silent blocks* interrupts the impact sound (body sound) as is usual in all passenger cars and trucks. However, it requires a frame chassis, which can increase initial costs. They can be balanced by additional benefits such as more flexibility of engine dimensions and makes or reduced added costs for interface elements of front implements and front end loaders (see John Deere family 6000, chapter 3).

Encapsulation of the engine is very effective in combination with silent blocks [6.40] but has limits regarding openings for combustion air and cooling, mainly for large tractors. The reason: *Engine power increases by at least a power of three of the tractor scale factor, but the tractor surface areas increase only by a power of two*. Air input speeds cannot be increased for compensation as there are limits on keeping away plant fibre elements from the input screens. The larger the tractor, the larger the sum of all openings for burning and cooling air in relation to the total bonnet area.

When the engine noise is under control, the noise of transmission and hydraulics becomes dominating. A combined suspension in silent blocks together with the engine was, for example, applied successfully in a special tractor [6.41] but is not used for standard tractors. In summary, silent blocks play a major role regarding "interruption" of the impact sound of tractor structures, **Fig. 6.31**.

Early cabs which were mounted rigidly on the tractor body had no sound reduction, the levels even increased [6.42]!

6.6.2 Dust, climate, and pesticides

Dust can be a problem when the tractor has no factory-installed cab. *W. Batel* reports dust concentrations near the tractor of between 1 and 500 mg/m^3 [6.43].

Other studies [6.44] yield typical values of 34–195 mg/m^3 for dry soil near the driver if there is no cab. They all recommend closed cabs with *microfiltered pressurized air entrances* achieving concentrations of only 0.03–0.50 mg/m^3.

Performance levels of various filter types are listed in [6.45], including those for microorganisms, see also ISO 179454-1.

An overpressure of 30 Pa was recommended for closed cabs in [6.46], while at least 30 m^3/h are stated in [6.4] to replace the air inside the cab**.**

Climate addresses *temperature, humidity, air intake, and air speed***.** Comfort figures in **Table 6.10** [6.47] represent a guideline for practical evaluations.

Detailed requirements are specified in **Table 6.11**, based on ISO 14369-2 (filter test in ISO 17954). ISO recommends an air intake flow of at least 43 m^3/h per person at a pressurization of at least 50 Pa. EN 15695 requires 30 m^3/h as a must. Without any technical aids, the temperatures in a cab can reach 10–15 K above those outside the cab in full sunshine. A blower can reduce this difference to 6–10 K; however, only at very high air input rates of 200–400 m^3/h.

Air conditioning systems are able to keep indoor temperatures considerably below outdoor values. Table 6.11 refers to comfort zones as defined in ISO 14269-2. The shaded area of **Fig. 6.32** shows a recommended zone of optimal comfort between 24 and 27°C, relevant for air conditioning systems.

Table 6.10: Guideline for driver's climate comfort [6.47]. For detailed updated values, see relevant ISO standards

Human feeling:	comfortable	not comfort.	unacceptable
Air input, m^3/h	30–90	10–30	below 10
Air temperature (driver's head), °C	W 18–20*) S 24–27	0–18 27–37	below 0 above 37
Rel. humidity, %	30–65	W 15–30 S 65–85	W below 15 S above 85
Air speed, m/s	0.1–0.5	0.5–1.0	above 1.0

*) W = Winter, S = Summer

Table 6.11: Recommended climate data for cabs

Subject	Recommendation of ISO 14269-2
Fresh air	
Filtered fresh air input	> 43 m^3/h (at max. setting)
Air pressurization	> 50 Pa (max. 200 Pa)
Air filter efficiency	> 96% (w/defined test dust)
Air velocity	< 0.3 m/s (at driver's eyes)
Cooling	
Temperature target	Comfort zone or 11 K below ambient temp. of > 38 °C
Ambient temp. at test	32 °C dry bulb 25° C wet bulb
Engine power at test	> 50%
Tractor speed	< 5 m/s
Heating	
Temperature target	Comfort zone or 36 K above ambient temp. (< -12 °C)
Ambient temp. at test	< -7° C (tractor cold)

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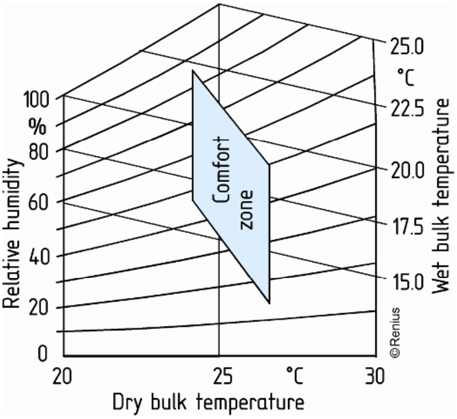

Fig. 6.32: Comfort zone for the driver in tractor cabs, relevant for cooling in summer with air conditioning systems. Source: ISO 14269-2, modified

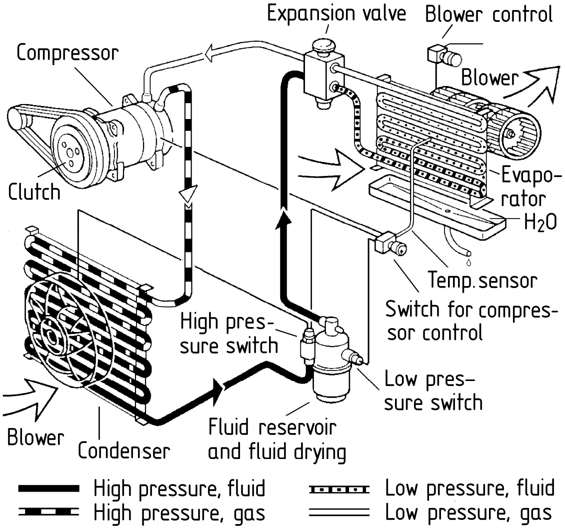

Fig. 6.33: Air conditioning system as used for vehicles, moving from right to left. Courtesy Behr (modified)

Air conditioning systems work like refrigerators by using special fluids with low evaporating temperatures in a closed circuit, **Fig. 6.33**.

The gaseous refrigerant is compacted by the compressor to, for example, 16 bar, which in turn increases its temperature to about 100°C. The pressurized gas moves to the condenser, in which the gas changes state into a liquid when cooled. When it arrives at the expansion valve, the refrigerant is throttled, reducing its pressure. The fluid evaporates, requires a very high amount of specific energy and changes its state into cold gas. This flows through the evaporator that cools the cab intake air.

Air conditioning increases the driver's comfort *and performance*. A compressor power of 3.3 kW was measured for a cab temperature of 8 K below outside in full sunshine [6.48]. *Göhlich* cites 2–4 kW as an average for similar conditions [6.4]. This needs about 0.5 to 1.0 liter of extra diesel fuel per hour. Design rules for energy savings are listed in **Table 6.12**.

Table 6.12: Design rules for keeping the energy consumption of air conditioning low

- Cab roof overhang producing shadow
- Cab with tinted glass
- Thermal insulation of the cab
- Favorable (cool) position of air intake
- Favorable (cool) position of condenser
- Insulated pipes for refrigerant
- Compressor only active on demand
- Compressor with infinitely variable speed
- Effective heat exchangers
- Refrigerant fluid with high cooling effect
- Energy-efficient control system

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Evaporative cooling systems for cabs cost much less, perform less well but are of interest for small tractors in developing countries, proposed with solar cells [6.49] to drive the cooling fan and the water pump.

Spraying pesticides can be a problem as conventional dust filters cannot retain chemicals. European standards [6.50, 6.51] define a cab classification in four levels. The highest must be able to retain pesticides. This class is obligatory within the EU for tractors working with spraying implements.

Fig. 6.34 shows an air flow and pressure situation. Special cab filters are on the market. Air leakage must be kept low.

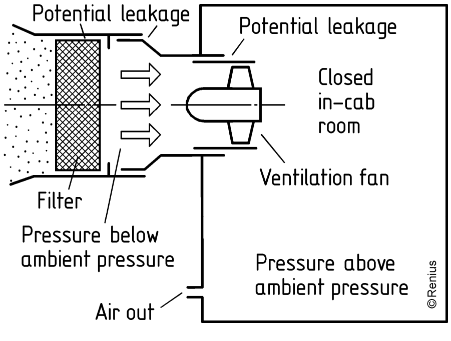
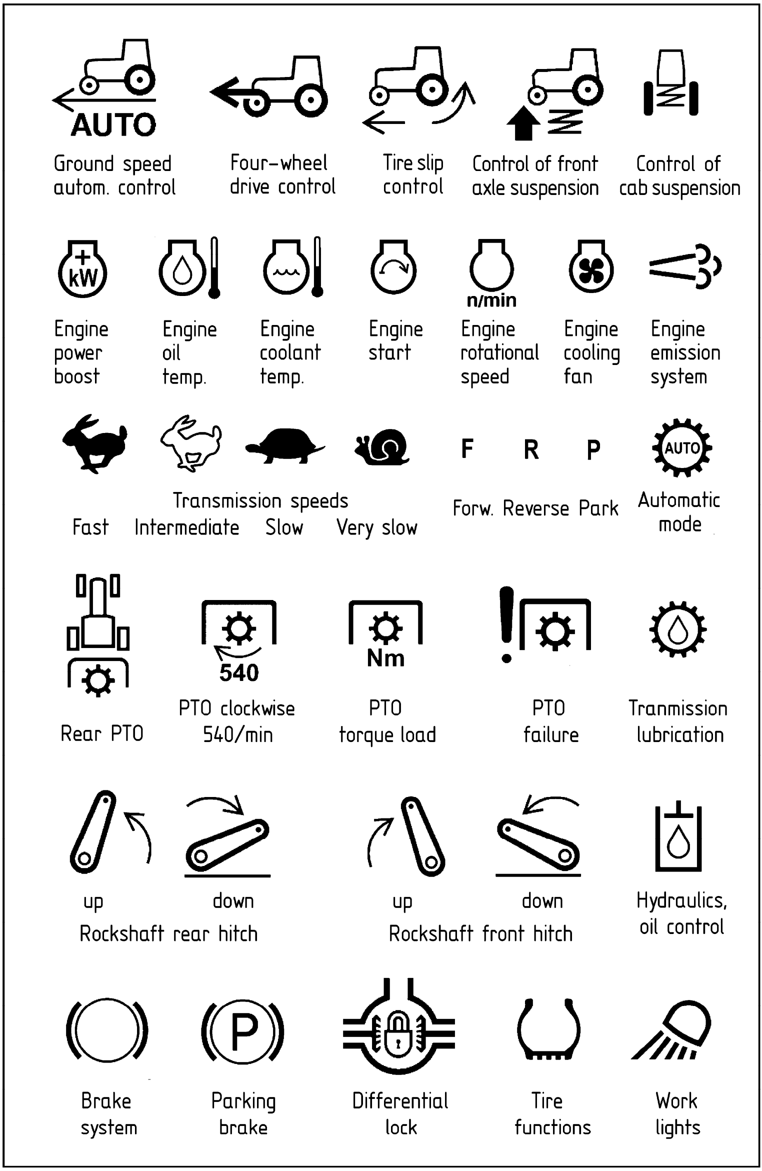


Fig. 6.34: Tractor cab: air flow and pressures

6.6.3 International symbols

International symbols improve orientation for the driver at the workplace with respect to levers, pedals, dashboards, terminals, and safety warnings, **Table 6.13**. Basic symbols which are well known from passenger cars and trucks are not included in this survey (to save space).

Table 6.13: Symbols for operator controls and displays, based on ISO 3767-2. Safety signs and hazard pictorials see ISO 11684:1995 (in 2019 under revision)



6.6.4 Tractor overturn safety

Fatal overturn accidents became a real problem with the general introduction of tractors in agriculture. As reported in [6.52], some 30,000 lives were lost in the US from 1921 to 1971 owing to overturning.

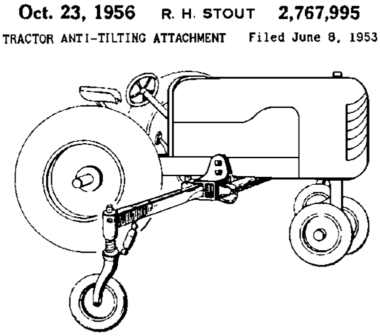

Oct. 23, 1956 R. H. STOUT 2,767,995
TRACTOR ANTI-TILTING ATTACHMENT Filed June 8, 1953

Fig. 6.35: US Patent 2,767,995 A on a structure preventing sideways overturning by telescoping side bars, see also Fig. 1.13

Table 6.14: Introduction of ROPS, milestones

1954	Early studies of *H. Moberg*, Sweden
1956	Early ROPS were tested at the University of California, Davis, USA
1957	First proposal of a pendulum test by *Moberg*
1959	Roll-over Protective Structures (ROPS) obligatory for newly registered tractors in Sweden
1960	US patent 3,455,598 by J. Deere, charge-free offered to competitive tractor manufacturers
1966	OECD Test Code 3 for dynamic ROPS test
1966	John Deere offers ROPS for all tractors
1967	ASAE Standard R 306 (USA)
1967	Pendulum test part of OECD Test
1969	In-depth research in USA, UK (NIAE), and Germany (TU Munich)
1970	ROPS obligatory in Germ. for all new tractors
1972	John Deere Sound Guard Cab in USA with integrated ROPS (Nebraska Test 1110, 1972)
1977	EC 77/536 defines dynamic ROPS Test
1977	ROPS obligatory in Germany for complete tractor fleet (≈ 1 M). All ROPS calculated and examined at TU Munich (Schwanghart/Söhne)
1979	EC 79/622 defines static ROPS test
1981	ISO 3463 (dyn.) and ISO 5700 (stat. test)
1983	Static tests are becoming generally used
1986	ASAE designates two locations as national historic landmarks of agricultral engineering for ROPS: Univ. of Calif. Davis and J. Deere

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Many proposals were made to solve this problem through warning or training the driver for improved safety sensitivity.

Sideways overturns were clearly dominating. A funny US patent, filed in 1953, proposed a safety side bar, **Fig. 6.35**. A commercial success is not reported.

The proposal of "**R**oll-**O**ver **P**rotective **S**tructures" (**ROPS**) promised much more success. A comprehensive study on their history was presented along with the *Historical Landmark Dedication Ceremony* by ASAE on Sept. 25, 1986 [6.53].

Early studies with various protecting structures were carried out, particularly in the US and in Sweden, **Table 6.14**.

Several patents were applied [6.53] but factory-installed safety structures did not become popular for a long time.

Sweden was the first country to introduce tested ROPS, becoming obligatory for new tractors [6.54]. The test was carried out by using a pendulum, **Fig. 6.36**. This method became widely accepted, Table 6.14, being cheaper and more easily repeatable than field overturns.

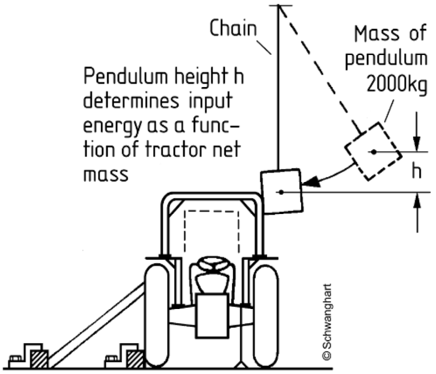

Chain
Mass of pendulum 2000kg
Pendulum height h determines input energy as a function of tractor net mass
h
Tractor rigidly fixed to the ground
©Schwanghart

Fig. 6.36: Testing ROPS or complete cabs using a 2000 kg pendulum from the side and the rear. Test energies are derived from the tractor mass and realized by the height h.
Deflection of the structure is limited by a defined clearance zone simulating the driver

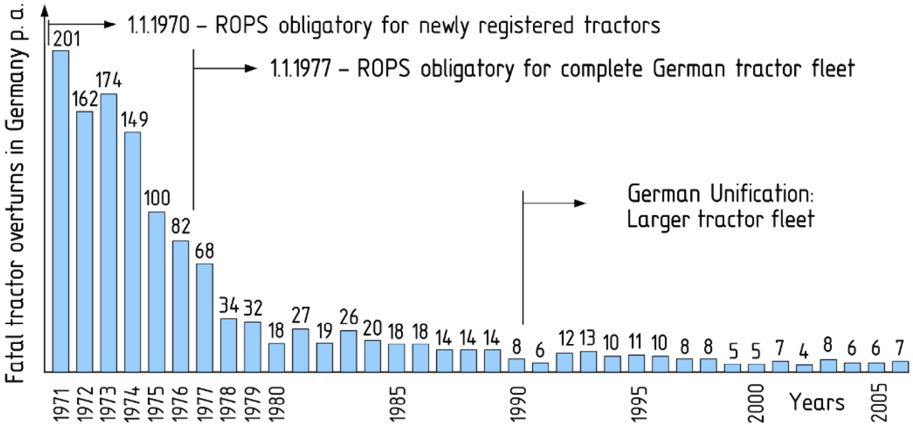
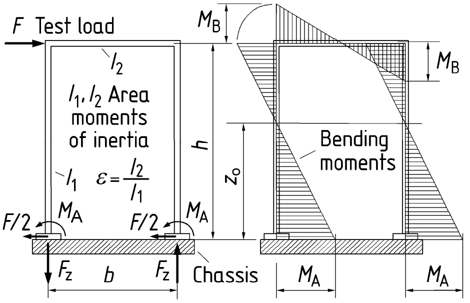


Fig. 6.37: Fatal overturns in Germany, reduction from about 15 per 100,000 tractors and year to under 1 per 100,000 tractors and year. Main influence from obligatory introduced ROPS

Research was carried out in Sweden by *H. Moberg* [6.54], in the US at the University of California, [6.52], at John Deere in Waterloo [6.53], at Silsoe Research Institute, UK [6.55], and in other places such as Italy and Germany [6.56, 6.57].

In 1969, 181 lives were lost in Germany due to overturning accidents. The Agricultural Workers' Insurance (BLB) became active and achieved that ROPS were obligatory from Jan. 1, 1970 for newly registered tractors, **Fig. 6.37**. The great success prompted the decision to equip the total German tractor fleet of about one million tractors with ROPS by Jan. 1, 1977.



$$z_o = h \cdot \frac{3\varepsilon \cdot h/b + 1}{6\varepsilon \cdot h/b + 1} \qquad M_A = F/2 \cdot z_o \qquad M_B = F/2\,(z_o - h)$$

Fig. 6.38: Basic mechanics of a simple frame dealing as a protective structure. Side force *F* simulates a sideways overturning load [6.57]

Such a high number of ROPS could not be tested. *Söhne* and *Schwanghart* developed a computer program at TU Munich evaluating all the designed ROPS for the complete tractor fleet [6.56, 6.57].

In order to get a feeling for such calculations, **Fig. 6.38** shows bending moments deforming a simple structure, first elastically, than plastically, both absorbing the required test energy. As demonstrated in **Fig. 6.39**, full section profiles are able to do this most effectively. Similar curves are registered for ROPS or complete cabs.

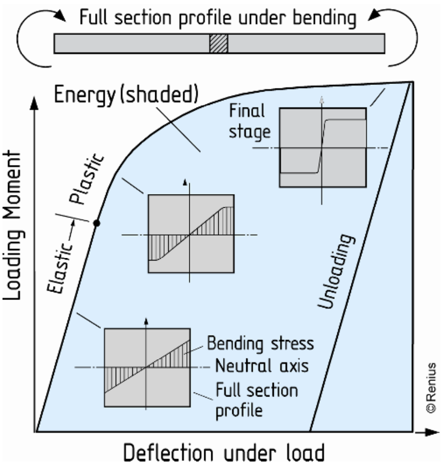


Fig. 6.39: Absorbed energy of a ROPS profile

ROPS design and test. Mechanics of sideways tractor overturning and critical typical slopes were presented in chapter 3.

The first **OECD Code 3** (introduced in 1966) specified the dynamic ROPS test for wheeled two-axle standard tractors using a 2 t pendulum with the following conditions:

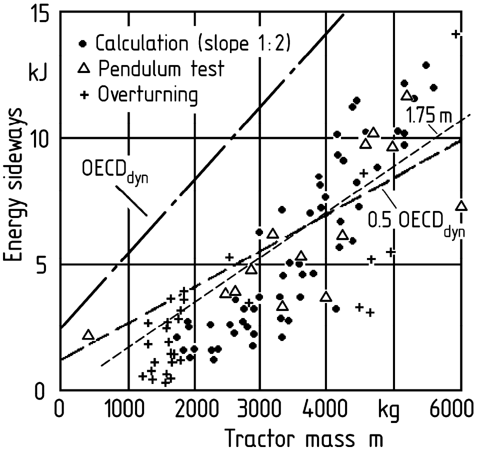
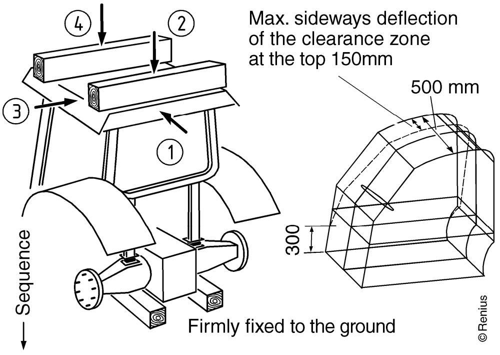


Fig. 6.40: Energy input to the ROPS from the side, derived from calculation, pendulum tests and practical overturn analysis as investigated at TUM by *H. Schwanghart* [6.58].
Static energy to be absorbed by the ROPS (OECD Code 4) is introduced as line 1.75 m


④
②
③
①
Max. sideways deflection of the clearance zone at the top 150mm
500 mm
300
Firmly fixed to the ground
©Renius
Sequence

1 Longitudinal rear ..... 1.4 m
2 Vertical rear .... 2.0 m
3 Sideways .... 1.75 m
4 Vertical front .... 2.0 m

Required energy in Joule = Nm.
m = unballasted tractor mass, kg

Fig. 6.41: Static ROPS test specifications as defined in OECD Code 4 and ISO 5700

Unballasted tractor mass between 600 and 6000 kg and a minimum track width of the rear wheels of 1150 mm.

As tractor power increased, net tractor mass exceeded the 6000 kg limit. This problem was solved by the development of the static ROPS test of **OECD Code 4**, introduced in 1983.

Specifications had been prepared by broad research to find out those static test energies which harmonize with the ROPS energy as absorbed by the pendulum tests.

One researcher was *H. Schwanghart* (under *W. Söhne* at TU Munich), who derived such relationships for *sideways pendulum tests, practical overturns, and calculation results,* **Fig. 6.40** [6.57]. He concluded that only about half of the pendulum energy deforms the ROPS. He found the same relation for the rearward pendulum tests, however, with a broader statistical spread.

Research by *Chisholm* (N.I.A.E., UK) [6.59–6.61] also supported the OECD Code using a computer model validated by full-scale experiments. An EU Directive 77/536/EC for ROPS tests was introduced in 1977 following joint research by *Boyer* (France), *Chisholm* (UK), and *Schwanghart* (Germany) to establish energy criteria [6.62].

Meanwhile, the static test as defined in **OECD Code 4** [6.63] dominates, **Fig. 6.41**. One reason is its ability for large tractors. The sideways energy harmonizes with Fig. 6.40. Further publications see [6.64–6.67].

OECD Code 6 [6.68] specifies testing of front-mounted ROPS on narrow-track wheeled tractors.

OECD Code 7 [6.69] addresses rear-mounted ROPS of narrow-track wheel tractors while **OECD Code 8** [6.70] was developed for ROPS testing of track-laying tractors.

6.6.5 Visibility

When tractors are equipped with cabs, visibility for the driver is reduced and should be checked for three reasons:

- Meeting *legal requirements* when driving on public roads
- Good view for *performing field work*
- Offering safety for *people near the tractor;* for example, children

Visibility can be measured by using two point sources of intense light which are positioned inside the cab, representing the eyes of the driver (with respect to the seat index point), **Fig. 6.42**.

The procedure is standardized for the forward view in ISO 5721-1. A circle with a radius of 12 m is used to define tolerated masking effects. The most important free view section is in front measuring 9.5m in width. To meet this requirement, mufflers of modern tractors of the higher technology levels are usually positioned in the "shadow" of the right cab front post, as shown in **Fig. 6.43**.

A position of the muffler ending at the bottom of the tractor chassis also offers a free view but is not recommended, as the exhaust gas can recirculate.

Very important for row crop operations is the visibility of the front wheels from inside the cab, resulting in front glazing extending below the hood down to the cab bottom level, Fig. 6.43.

Side and rear vision requirements are standardized in ISO 5721-2.

Side fields of vision (left and right) are important to prevent collisions with persons, animals, or other vehicles, and are mainly important for large tractors with high rear tires, **Fig. 6.44**. Further requirements are settled for the rear field of vision.

In order to increase safety even beyond all these recommendations, cameras have been introduced to monitor non-visible sections; see, for example ISO 20112-1.

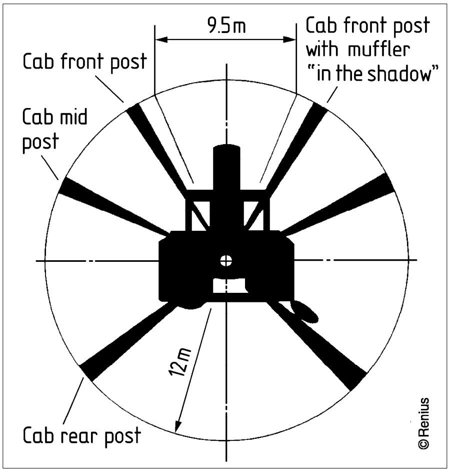


Fig. 6.42: Visibility diagram, shadow silhouette produced by two point sources of light representing the position of the driver's eyes. This example of a standard tractor documents a very good visibility. After *L. Magnus* 1980

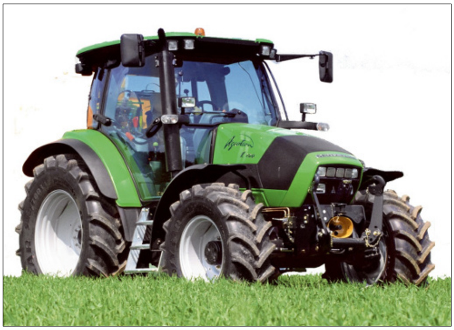


Fig. 6.43: Exhaust pipe in the "shadow" of the right cab front post. Courtesy Deutz-Fahr

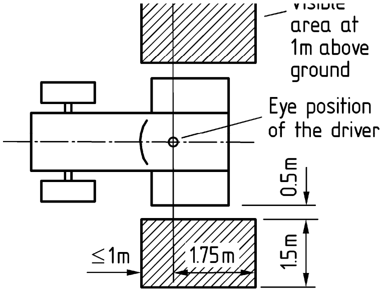


Fig. 6.44: Required side visibility in ISO 5721-2

6.6.6 Completely noise- and dust-insulated safety cabs

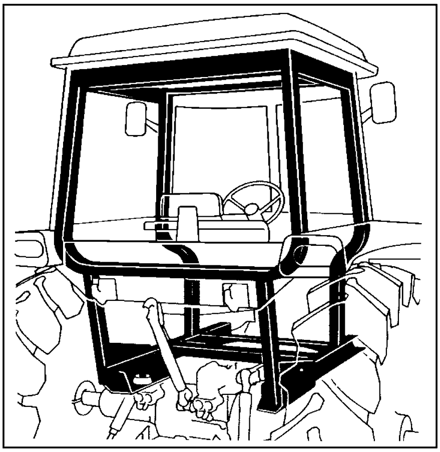


Fig. 6.45: Safety structure of the John Deere "Sound-Guard" cab, 1972. Access from the left door at the front. Courtesy John Deere

It was already mentioned in the beginning of chapter 6 that John Deere took an important step with its "Sound-Guard" offering the first mass-produced noise- and dust-isolated cab for standard tractors with an integrated safety structure, **Fig. 6.45**.

The (black) safety frame was put on the tractor chassis via silent blocks similar to those of Fig. 6.31. The structure is reinforced in its upper and lower parts owing to the high bending moments in these areas as explained in Fig. 6.38.

This cab was evaluated in Nebraska Test No. 1110 Oct. 5–13, 1972 on a John Deere 4430 reporting a noise level at the driver's ear of only 82.5 dB(A) under full power.

Other tractor and several cab manufacturers followed. The closed "quiet cab" soon became popular, **Fig. 6.46**.

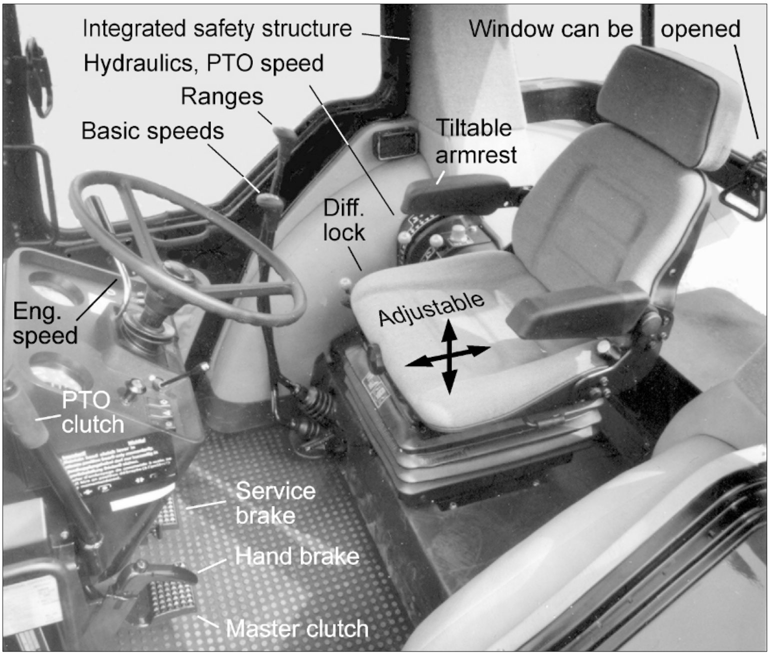


Fig. 6.46: Workplace inside a simple "quiet cab". View from the main cab access side. Access from the opposite side disabled by shift levers and foot pedals. Such a design was typical of the first safety cab generations 1975–1990, now typical for technology levels II–III. Courtesy

Large premium tractors now offer very modern cabs with complete drive-by-wire equipment, multifunctional armrests, and a large number of automatic functions.

Two examples are shown in **Fig. 6.47** and **Fig. 6.48**. Both offer a comfortable instructional seat (ISO 23205) and a good view of the front wheels as well.

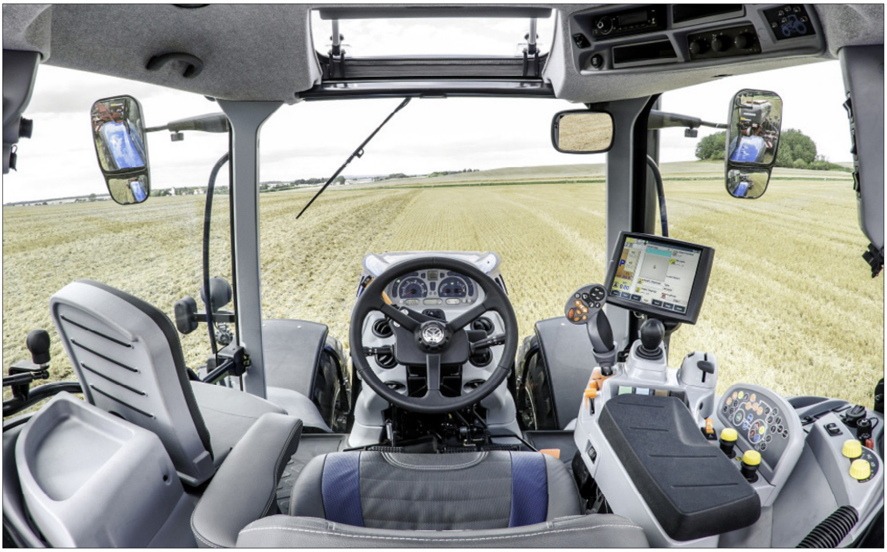


Fig. 6.47: Workplace of New Holland tractor line T7 as of 2018. Rated power ISO/ECE 120: 92–168 kW. Instructional seat (left), multifunctional armrest Sidewinder^{TM} II. Courtesy CNH

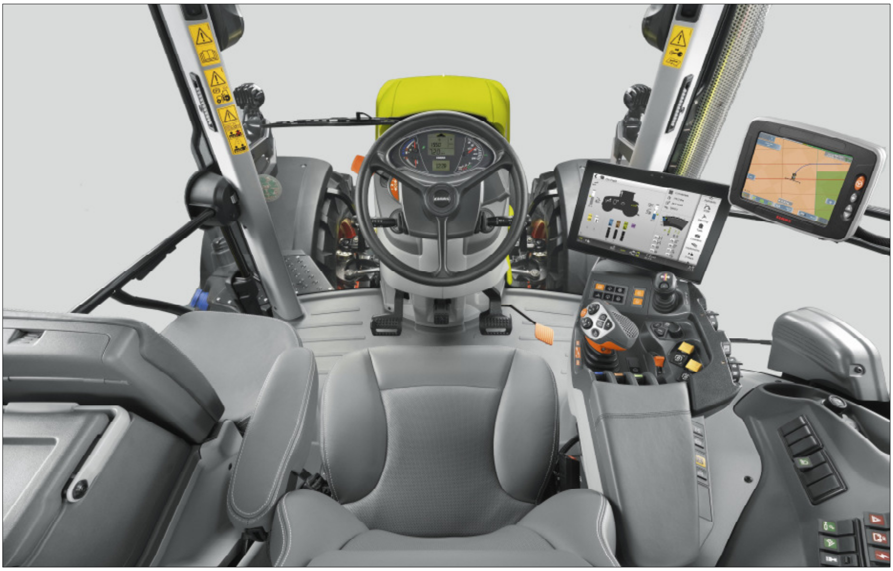


Fig. 6.48: Workplace of CLAAS tractor line Axion 800 (2018), rated engine power 142–199kW. Instructional seat (left), multifunctional armrest. Courtesy CLAAS

Four cab classes. Protection against chemicals was a problem for a longer period of time. As mentioned in chapter 6.6.2, standards EN 15695-1 and EN 15695-2 were developed which define *four classes of filter types according to the kind of protection,* **Table 6.15**.

Table 6.15: Tractor cab classes 1–4 according to its filter equipment, as standardized in Europe [6.50]. Test specifications see [6.51]

Cab class	Level of protection
1	Provides a specified level of protection against hazardous substances
2	Protects against dust
3	Protects against dust and aerosols
4	Protects against dust, aerosols, and vapours

Class 4 cabs are a must in many countries in which farmers spray chemicals.

Cabs of premium tractors offer a multifunctional armrest as a standard or option. The technical effort needed is very high as the seat structure must be able to carry the armrest with all its control elements and the signals must be transmitted through the movable sub frame by wires.

This is very costly and hence restricted to premium tractors, **Fig. 6.49–6.52**.

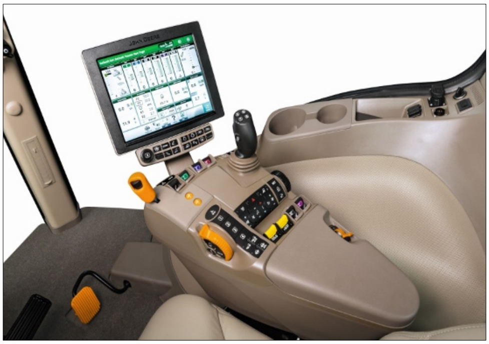


Fig. 6.49: Multifunctional Armrest of John Deere premium tractors, introduced with the 8000 line in 1994. Courtesy John Deere

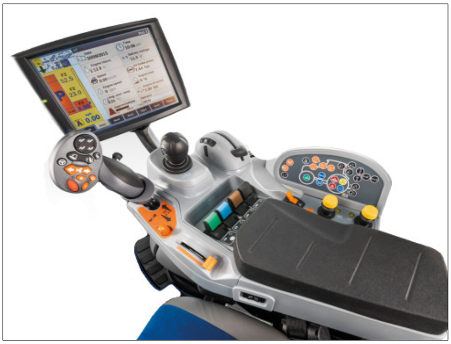


Fig. 6.50: Multifunctional armrest Sidewinder II, NH tractor line T7, 2018. Courtesy CNH

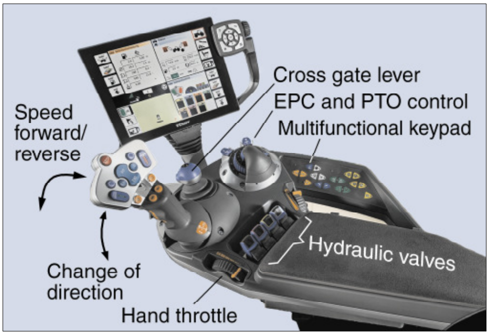


Fig. 6.51: Multifunctional armrest of Fendt Vario Terminal for premium tractors as of 2018. The first Vario Terminal generation was presented in 1998 and introduced in series in 1999 in the 700 Vario line. Courtesy AGCO-Fendt

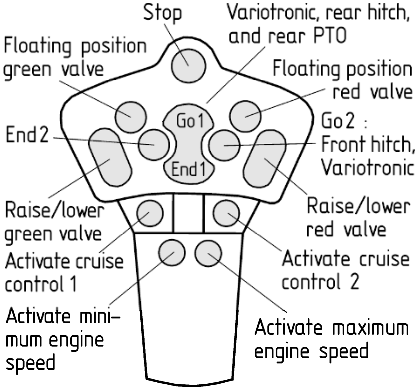

Stop
Variotronic, rear hitch, and rear PTO
Floating position green valve
Floating position red valve
Go1
End1
End2
Go2 : Front hitch, Variotronic
Raise/lower green valve
Raise/lower red valve
Activate cruise control 1
Activate cruise control 2
Activate minimum engine speed
Activate maximum engine speed

Fig. 6.52: Functions of the Fendt joystick of the Vario Terminal as of 2018. Drawing of the author, based on material from Fendt

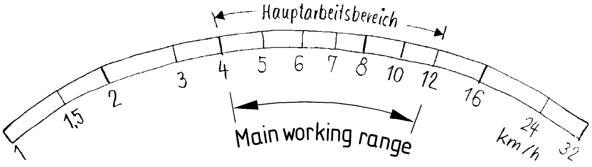


Fig. 6.53: Proposal of a logarithmic speedometer scale, original drawing of the author, 1980 [6.71]

Displays for engine and tractor speeds are today often digital. *Analogue speedometers offer in addition a survey of the actual speed in relation to the total range.*

A linear scale is sufficient for the engine speed but not useful for the vehicle speed as its bandwidth is much larger, ranging at least from about 1 to 32 km/h, resulting in poor readability of low speeds

This is the reason that the author proposed during his appointment at Deutz a logarithmic scale replacing the linear one, **Fig. 6.53** [5.71].

The relative accuracy of reading is now constant for the total range. The idea was not employed for lack of low-cost electronics.

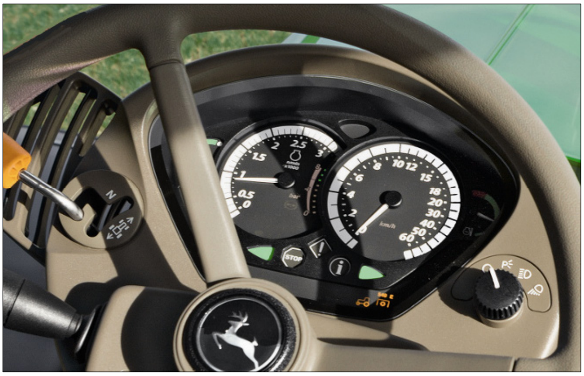


Fig. 6.54: A dashboard from John Deere. Courtesy J. Deere

It seems that John Deere was thinking in the same direction when scaling the analogue tractor speed display of its dashboards almost logarithmically, **Fig. 6.54**.

In addition to analogue speedometers, premium tractors feature digital speed displays on their monitors. They offer greater accuracy which is, for example, welcomed for cruise control.

The cab access of **Fig. 6.55**, meets the requirements of ISO 4252 (Fig. 6.7). A variable height of the lowest step can solve problems with different tire diameters. The cab doors are often frameless “only glass” with surrounding sealing and metal fittings, saving costs and weight. The picture also shows the typical position of the plastic tractor tank for fuel, utilizing any available space for adequate fuel capacity.

The slip-proof steps are protected from mud by special fender elements.

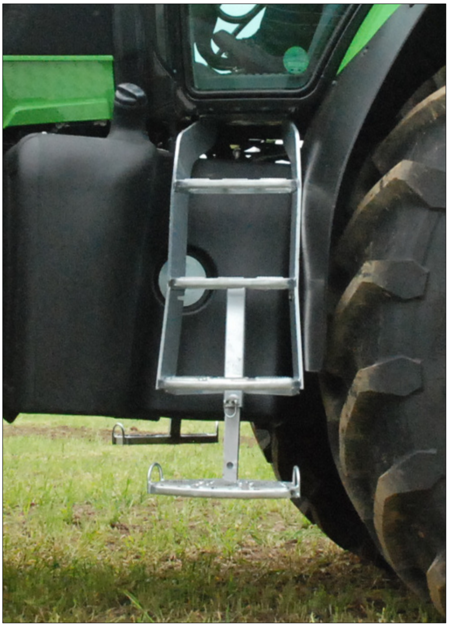


Fig. 6.55: Lowest step adjustable, fender prevents mud deposit (Deutz). Photo L. Volk

Safe mounting and dismounting. The introduction of integrated ROPS reduced the number of fatal overturns (chapter 6.6.4), with the consequence that other accidents now became more important in relation to the total number of all accidents.

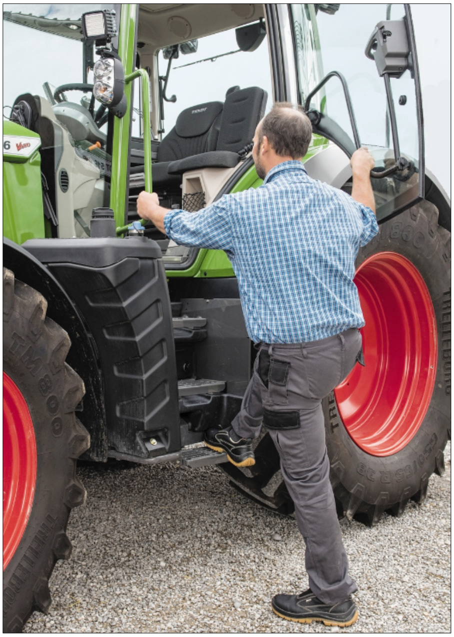


Fig. 6.56: Safe cab access and exit using three fix points. Courtesy AGCO-Fendt

This is, for example, the case for mounting and dismounting the cab. When getting on and off a tractor, the driver should always *keep three points of contact* (hand rails left and right, slip-proofed steps) with the tractor or ground, **Fig. 6.56**.

Soft suspensions for improved riding comfort became popular for cabs of premium tractors in addition to silent blocks.

In 1987, a first design was presented by Renault (France) with its TZ tractor line, at that time a progressive and popular approach, **Fig. 6.57**. The kinematics were already almost perfect due to longitudinal links, transversal wishbones, and a roll-bar stabilizer. A general breakthrough did not take place due to the high additional costs and the tractor height increase as well.

However, continuously increased tractor power led to higher total initial tractor costs with the result of decreased relative cab costs. Along with a trend towards more comfort and due to research results [6.4], soft cab suspensions became popular first in the high-power premium market segment, **Fig. 6.58**. The first structures (after Renault) were simple, but there are now also complex systems offered.

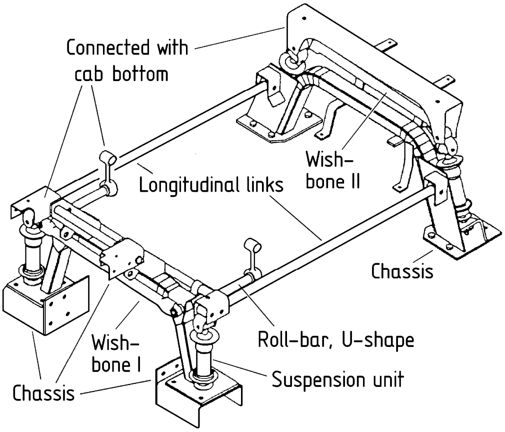


Fig. 6.57: First soft suspended cab (1987) for the Renault TZ tractor line. Courtesy Renault

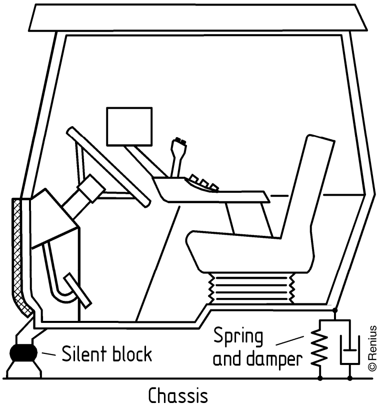


Fig. 6.58: Structure of a simple soft suspended tractor cab. Ideal vertical motion is slightly disrupted by additional rotation

There is a very broad bandwidth of **technical cab specifications** depending on local market demands.

Table 6.16 shows two typical cases: The *lo-tech cab* protects mainly against weather, noise, dust, and overturning and still contains several mechanical controls; however, observing impact sound. Specifications are typical for tech level III.

The right column is typical for *cabs of large premium tractors* with power levels typically above 100 kW. Outstanding comfort results from excellent reduction of vibrations and from climate control.

Drive-by-wire technologies and CANBUS networks enable automatic functions and connectivities. High absolute initial costs are accepted due to moderate relative costs.

Table 6.16: Typical specifications of tractor cabs for two technology levels

Function	Lo-tech cab	Hi-tech cab
Common functions	Integrated ROPS, power steering, air intake blower and dust filter (easy to replace), flat structured floor “waterproof”, “hanging” pedals, visibility of front wheels and rear hitch points, anchor points for seat belts (ISO 3776), tiltable front and rear windows, box for documents, all dimensions (including access and exit) meeting ISO standards, steps with grip, mud guard able to keep them clean.	
Max. noise @ driver’s ear	78–84 dB(A)	69–76 dB(A)
Seat, air suspended	Simple	High comfort
Steering wheel adjustment	Simple	Convenient
Right door of the cab	Closed	Can be opened
Air conditioning	Option	Standard
Cooling box for drinks	Option	Standard
Filter for chemicals	Option	Often standard
Multifunctional armrest	Not available	Often standard
ISO 11783 equipment	Not available	Standard
Large touchsreen	Not available	Option
Seat belts	Option	Standard
Manual gear shifting	Mechanical	By wire
Automatic gear shifting	Option	Standard
Engine speed control	Mechanical	By wire
Differential lock shifting	Pedal	By wire
PTO shifting	Lever	By wire
Remote control valves	Mechanical	By wire
External PTO & hitch control	Option	Standard
Automatic steering interface	Option	Standard
Automatic headland turns	Not available	Option
Instructional seat, foldable	Simple, option	Convenient, option
Soft suspended cab	Simple, option	Standard
Work lights	Option	Standard
Integrated radio	Option	Standard
Battery switch disconnector	Option	Standard
Window heating (front, rear)	Not available	Option
Infrastructure for smartphone	Simple, option	Standard

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6.6.7 Electronics and electrics

Electronic systems have been introduced more and more for premium tractors since about 1975, **Table 6.17**, see also CIGR Book, Vol. VI [6.72].

The first BUS systems were standardized in Germany based on proposals of *H. Auernhammer*, **Fig. 6.59**.

An internal bus for tractor components is connected with an open implement bus. Early applications are summarized in **Table 6.18**.

Table 6.17: Milestones of tractor electronics

1975	First electronic tractor control systems
1987	First BUS proposals of *H. Auernhammer*
1989	German standard DIN 9684, part 1, evolved to a final five parts
1991	Working group ISO TC23/SC19/WG1 starts work on ISO 11783, based on DIN 9684
2000	11 parts of ISO 11783 ready, some still as drafts; broad introduction begins
2006	Foundation of AEF (Agric. Industry Electronics Foundation) supporting ISOBUS
2013	ISO 11783 available in 14 parts

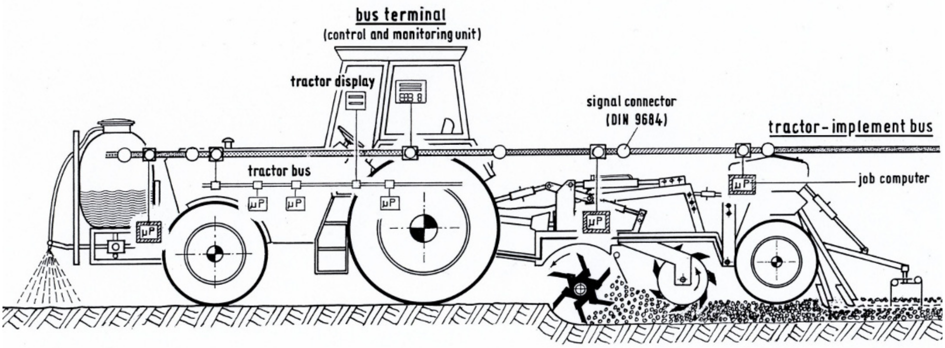


Fig. 6.59: German LBS (**L**andwirtschaftliches **B**US-**S**ystem): *H. Auernhammer´s* first drawing from July 14, 1987 [6.73, 6.74]. This structure became the basis not only for the German DIN 9684 (five parts) [6.75] but also for the international standard ISO 11783 ("ISOBUS"), which was expanded during 1991–2013 to a total of 14 parts (this was still the status in 2019)

Table 6.18: Tractor electronics, survey based on proposals of *H. Auernhammer*. The original publication from 1988 [6.76] contains an even more detailed breakdown

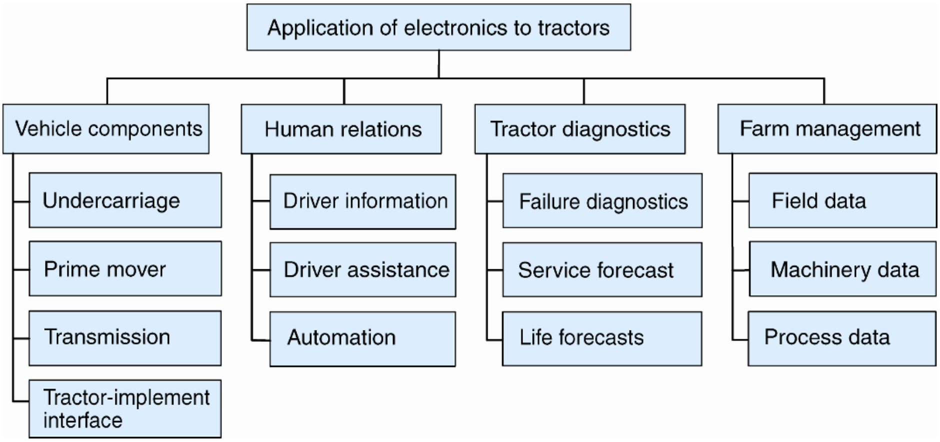
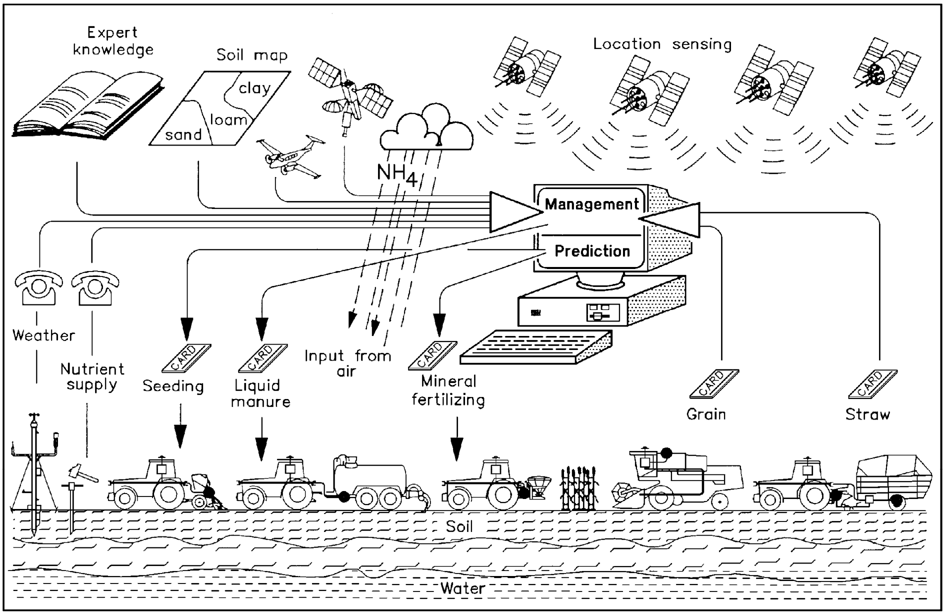


Table 6.19: Integrated precision farming, vision of *Auernhammer* and *Schueller* from 1999 [6.78]



In 1991, an ISO working group started development of ISO 11783 [6.77].

In 1999, the importance of electronic systems for "Precision Farming" was demonstrated in [6.78] by **Table 6.19**. This vision has meanwhile become reality with even more elements; for example, diagnostics, tele services, smartphone use [6.79] etc. The increased CANBUS applications can be read from **Fig. 6.60**.

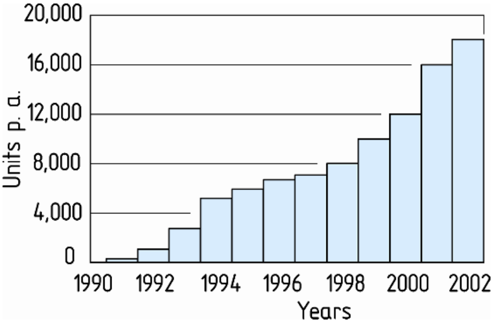


Fig. 6.60: Delivered CAN controller units of STW. Courtesy Sensor Technik Wiedemann (STW, DE)

The international standard ISO 11783 was developed in 14 parts, **Table 6.20**.

Table 6.20: ISO 11783, fourteen parts [6.80]

ISO 11783: Tractors and machinery for agriculture and forestry – Serial control and communications data network, as of 2013 (14 parts, still valid in 2018)	
ISO 11783-1	General standard for mobile data communication
ISO 11783-2	Physical layer
ISO 11783-3	Data link layer
ISO 11783-4	Network layer
ISO 11783-5	Network management
ISO 11783-6	Virtual terminal
ISO 11783-7	Implement messages application layer
ISO 11783-8	Power train messages
ISO 11783-9	Tractor ECU
ISO 11783-10	Task controller and management information system data interchange
ISO 11783-11	Mobile data element dictionary
ISO 11783-12	Diagnostics services
ISO 11783-13	File server
ISO 11783-14	Sequence control

Table 6.21: Example of a network according to ISO 11783-1, pins 1–9 are defined in Table 6.22. Regarding the tractor ECUs, there is a trend to integrate them into a central package

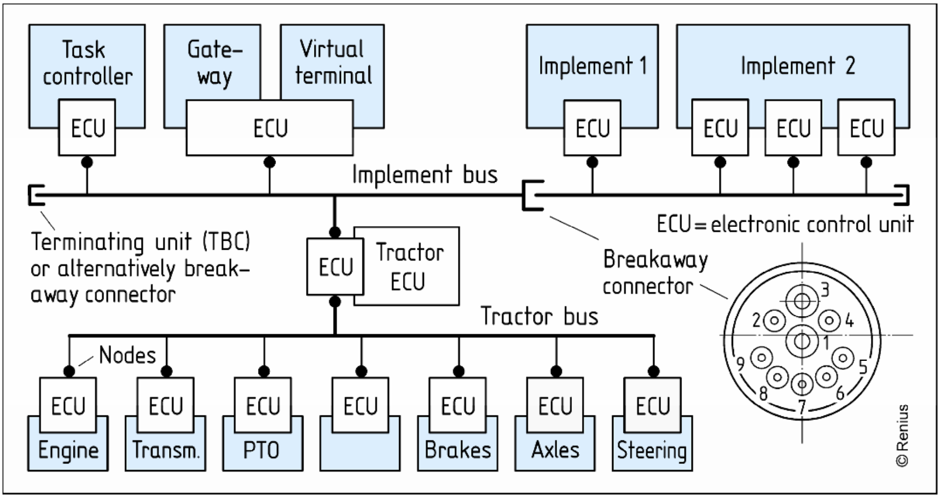


Table 6.22: Pin allocations of Table 6.21 according to ISO 11783-2, first issue 2002 still valid

Pin and code	Colour	Function
1 GND	Black	Ground tractor and impl., ≤ 60 A
2 ECU_GND	Black	Isolated impl. ground, ≤ 25 A
3 PWR	Red	12 V power for implements, ≤ 60 A
4 ECU_PWR	Red	12 V power for impl. ECUs, ≤ 25 A
5 TBC_DIS	-	Control of terminating bridge
6 TBC_PWR	Red	12 V power for BUS termination
7 TBC_RTN	Black	Ground for BUS termination
8 CAN_H	Yellow	Data transfer: Signal of CAN Hi
9 CAN_L	Green	Data transfer: Signal of CAN Lo

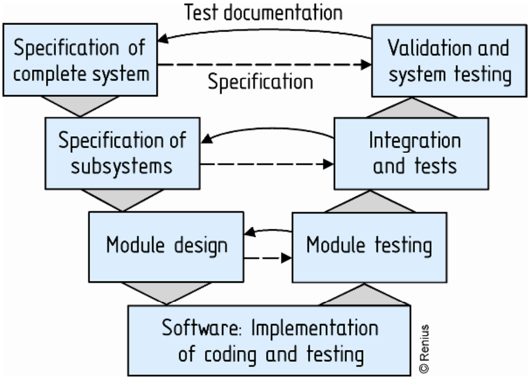

Test documentation
Specification of complete system
Specification
Validation and system testing
Specification of subsystems
Integration and tests
Module design
Module testing
Software: Implementation of coding and testing
© Renius

Fig. 6.61: V-model, originally proposed for software developments, since about 2004 also for mechatronic systems

The structure of ISO 11783 is illustrated in **Table 6.21** by a typical configuration, meeting ISO 11783-2: The principle of the two combined CAN BUS systems was originally also part of the German DIN 9684.

The pin allocation is explained in **Table 6.22** completing Table 6.21. The “implement breakaway connector” contains the *BUS signals hi and lo* for communication and also pins for electric power transfer for which ISO 11783 also specifies wire cross sections.

Systematic development of electronics can be structured by using the V-model, **Fig. 6.61** [6.81].

*Developing **safety measures** as early as possible along with the software has been proved* [6.81].

ISO 12100 and IEC 61508 have defined general safety standards for machinery electronics since 2010.

ISO 18497.2 addresses safety of highly automated agricultural machinery, hence includes tractors.

Research at TU Munich [6.81–6.84] addressed innovations through electronics. This supported, for example, the development of “AGRO NAV” (from GEO TEC), the first commercially available automatic navigation [6.85]. Integration into a Deutz tractor was realized at the author’s former institute at TUM-Garching, modifying the hydrostatic steering system for electronic control. In 1999, GEO TEC received a DLG Gold Medal, the highest award of AGRITECHNICA’99, **Fig. 6.62**.

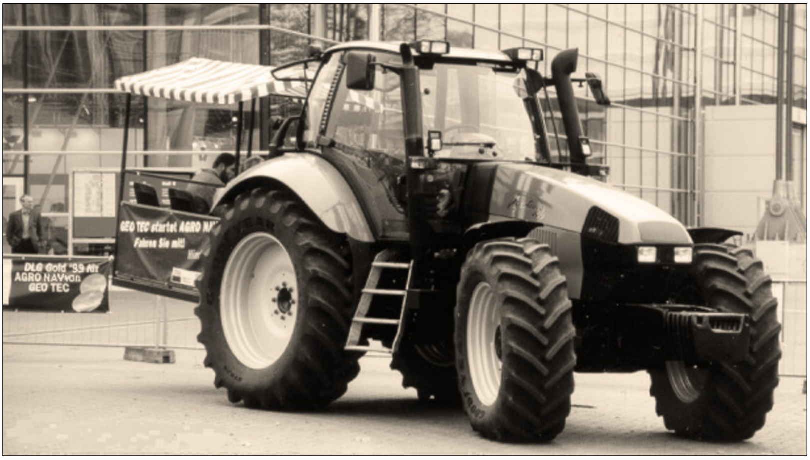


Fig. 6.62: First commercially available auto navigation: AGRO NAV from GEO TEC 1999, demonstrated by a Deutz tractor transporting people at AGRITECHNICA’99 without driver. © TUM

This first system from GEO TEC had a statistical sideways standard deviation, measured at the author’s former institute, of only about one inch but did not achieve comercial success due to its high price.

However, similar auto navigation systems later on became very popular for premium tractors, offered in graded precision levels (and related graded prices).

High accuracy, **Fig. 6.63**, requires corrected GPS signals [6.86].

In 2008, the AEF (**A**gricultural Industry **E**lectronics **F**oundation) was founded to support ISO 11783, mainly through conformance tests initiated by the industry, but also through public tests, for example through the *plugfests* which became very popular events [6.87].

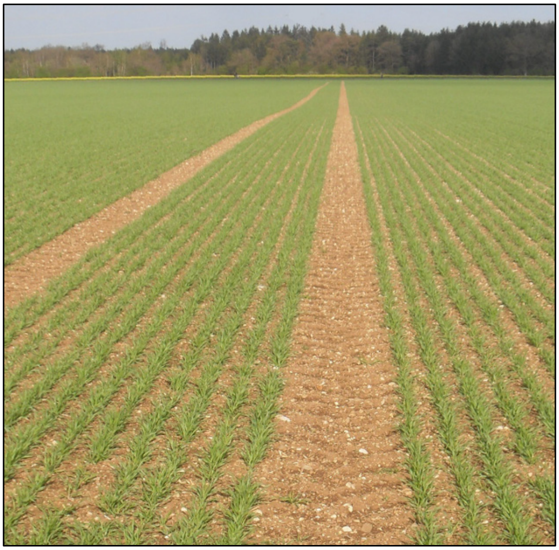


Fig. 6.63: Typical field in springtime in 2017 near Munich, Germany, demonstrating the precision of modern automatic tractor guidance systems. Photo: Author

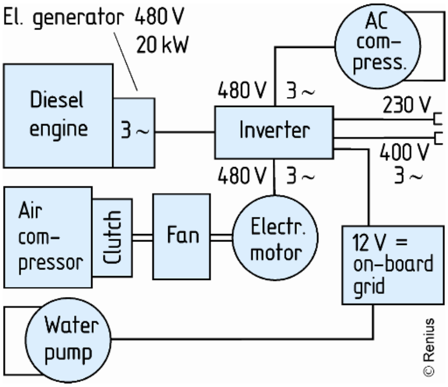


Fig. 6.64: First integrated electric power net (20 kW), presented by John Deere 2008 with its E-Premium tractors 7430 and 7530 (118 and 129 kW ECE R24)

Electrification of tractor functions has advantages in specific cases**,** mainly for increasing energy efficiency.

All larger tractor manufacturers investigated the potential. Diesel-electric ground drives were found to offer no sufficient benefits. However, there is potential on a lower power level to replace, for example, mechanical or hydrostatic drives, usually by electric 3-phase motors. Their output speed is controlled by frequency of inverters.

In 2008, a first series-produced system was introduced by John Deere, **Fig. 6.64**.

Premium tractors are today “filled” with a complex system of electrics and electronics, **Fig. 6.65**.

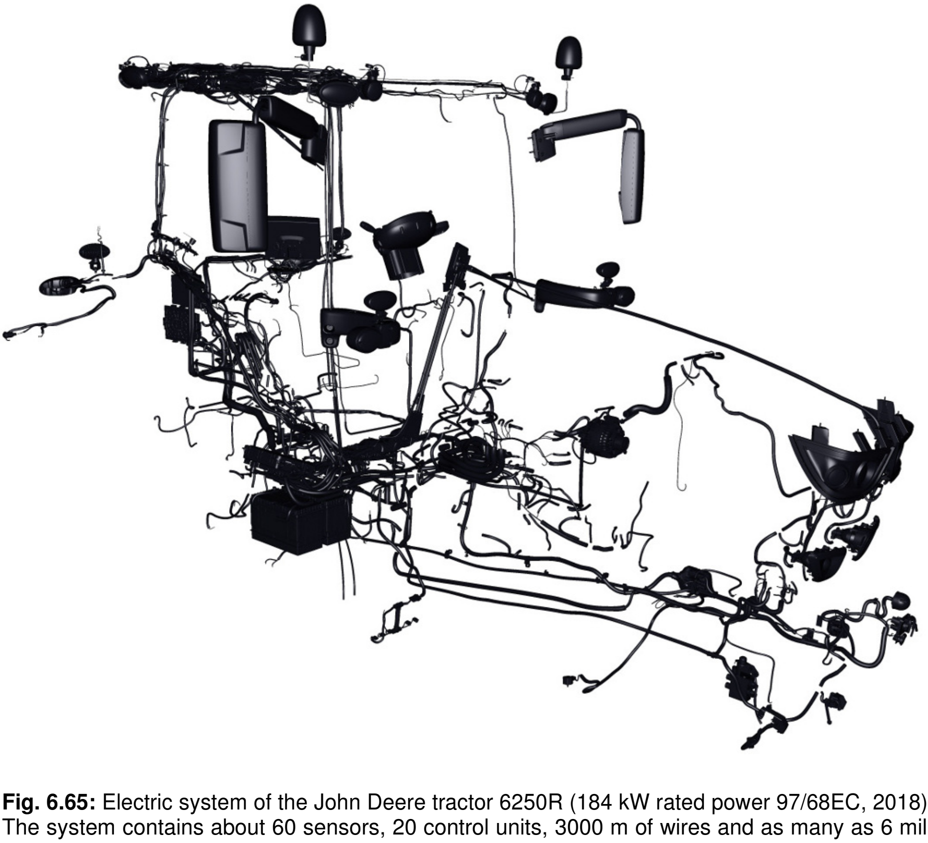


Fig. 6.65: Electric system of the John Deere tractor 6250R (184 kW rated power 97/68EC, 2018). The system contains about 60 sensors, 20 control units, 3000 m of wires and as many as 6 million lines of software. A disconnect master switch can prevent break downs of the battery due to leakage currents or not deactivated electric loads. Courtesy John Deere

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7 Tractor and implement

7.1 Introduction

The standard tractor becomes productive in agriculture only in combination with implements or trailers. A very few special tractors offer platforms for transport.

Used in a broad variety of configurations, tractor-implement combinations are the most important technical aids of agricultural mechanization; some typical versions for upland farming are shown in **Table 7.1**.

Tractors for wetland farming are often combined with only a rear rotary power tiller.

Basic requirements of tractor-implement combinations are listed in **Table 7.2**. They include public traffic law requirements, which are, for example, strict in Western Europe and similar regions. Demands of precision agriculture and smart farming are met by a high level of information and automation technologies.

Working processes can be improved considerably by replacing open loop controls by closed loop controls as already addressed in Table 6.2. The cybernetic principle of a closed loop control is demonstrated in **Fig. 7.1**:

The work output is continuously detected by sensors and compared to a given target in order to adjust the result permanently and automatically to the target by actuators.

An early, but very important example of applying this principle is the draft control for tractor-plow systems, see the famous British patents of *Harry Ferguson* (page 226) and the text below.

Advancements in *precision agriculture* are supported not only by the ISOBUS framework of the tractor-implement system (see chapter 6) but also by signals of global navigation satellite systems (e. g. GPS) which are improved by correction systems.

Table 7.1: Popular tractor-implement combinations

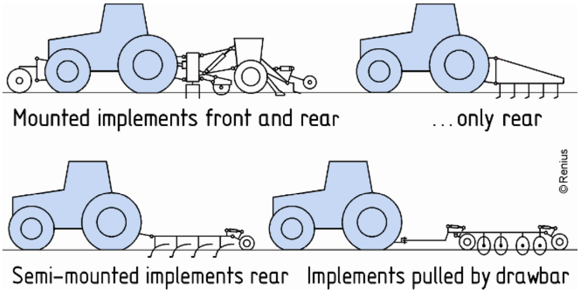
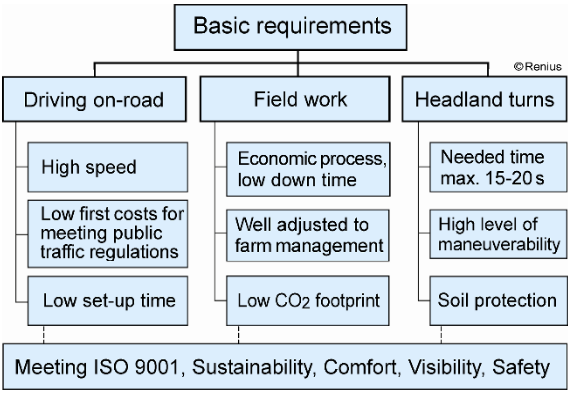

Mounted implements front and rear
...only rear
©Renius
Semi-mounted implements rear
Implements pulled by drawbar

Table 7.2: Tractor and implement: Basic requirements


Basic requirements
©Renius
Driving on-road
Field work
Headland turns
High speed
Economic process, low down time
Needed time max. 15-20 s
Low first costs for meeting public traffic regulations
Well adjusted to farm management
High level of maneuverability
Low set-up time
Low CO2 footprint
Soil protection
Meeting ISO 9001, Sustainability, Comfort, Visibility, Safety

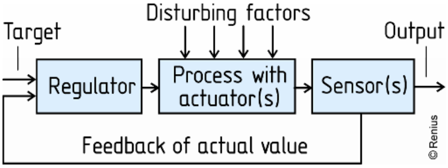


Fig. 7.1: Basic structure of a closed loop control system

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K. Th. Renius, *Fundamentals of Tractor Design*,
https://doi.org/10.1007/978-3-030-32804-7_7

Table 7.3: Typical sequence of actuations for headland turns of a system similar to that of Fig. 7.2

1 Move total control to headland process (GPS)
2 Raise front implement using tractor front hitch
3 Raise first rear implement using tractor rear hitch
4 De-activate rear PTO
5 Raise tool arm for track marker
6 Raise second rear implement using impl. hitch
7 Stop seeding process
8 Raise seeding machine using implement hitch
9 Disconnect differential lock of front axle
10 Disconnect differential lock of rear axle
11 Disconnect four-wheel drive
12 Reduce engine speed
13 Shift transmission to headland forward gear
14 Activate automatic steering
15 Increase and reduce engine speed
16 Shift transmission to headland reverse gear
17 Shift transmission to forward gear
18 Increase engine speed
19 Connect four-wheel drive
20 Engage differential locks
21 Lower front implement to working position
22 Lower first rear implement to working position
23 Activate rear PTO, selecting the right speed
24 Adjust engine speed for desired PTO speed
25 Put track marker in working position
26 Lower second rear implement to working position
27 Activate seeding process
28 Shift transmission to working speed
29 Move from headland to working control

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Complex implement combinations have become popular for premium tractors mainly in Europe, **Fig. 7.2** and **7.3**.

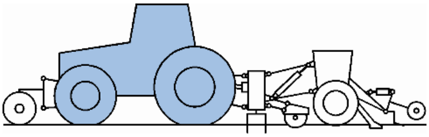


Fig. 7.2: Tractor-implement system for combined tillage and seeding

Field preparation and seeding can be done in only one path:

- Crack the clods at the surface in front
- Refine the soil with a power harrow
- Reduce the pores with a compactor
- Put the seed down in the right position
- Cover the seed by a light harrow.

Manual headland management is a very busy process as it consists of a high number of actuations, **Table 7.3**.

Automatic systems [6.83] are able to perform the complete sequence automatically in about 15 seconds, and meeting proposals of ISO 11783-14.

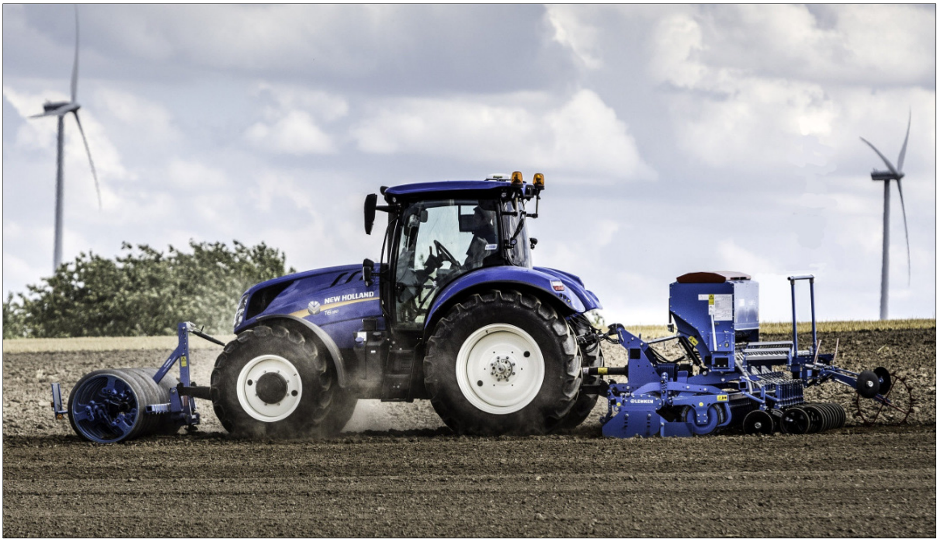


Fig. 7.3: Tractor-implement system for combined tillage and seeding in one path. Courtesy CNH

7.2 Tractor-implement management (TIM)

Automation of a working process of tractor-implement systems needs digital communication using ISOBUS (chapter 6.6.7) with closed loop controls (Fig. 7.1).

Criteria are *performance, work quality, energy consumption, environmental protection, comfort, safety, and finally economy.*

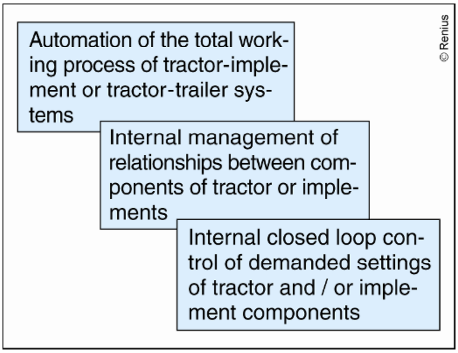
The author identifies three **main technical areas of automation**, **Table 7.4**, in which closed loop circuits play a central role. Communication between these levels is possible via ISOBUS; however, internal closed loop controls of components can work independently on the lowest level.

Examples for this lowest level are the controls of engine speed, transmission ratio, PTO speed, three-point hitch force and/or position, electric generator speed, air pressure system performance, and air conditioning compressor speed.

Examples for the mid level are the powertrain management of the engine-transmission system, engine-PTO powertrain, and engine cooling system.

The highest level of automation is called **Tractor-Implement Management** (TIM). It can, for example, be realized by the strategy "implement controls tractor," shown with two examples in **Fig. 7.4** and **7.5**.

Table 7.4: Three levels of automation



Automation of the total working process of tractor-implement or tractor-trailer systems

Internal management of relationships between components of tractor or implements

Internal closed loop control of demanded settings of tractor and / or implement components

Fig. 7.4: Baler sensors detect throughput, keeping it constant by adjusting tractor speed for optimum baler workload. Closed loop control via ISOBUS reduces the risk of material blockage and optimizes performance. Prototype stage in 2018. Courtesy AGCO-Fendt

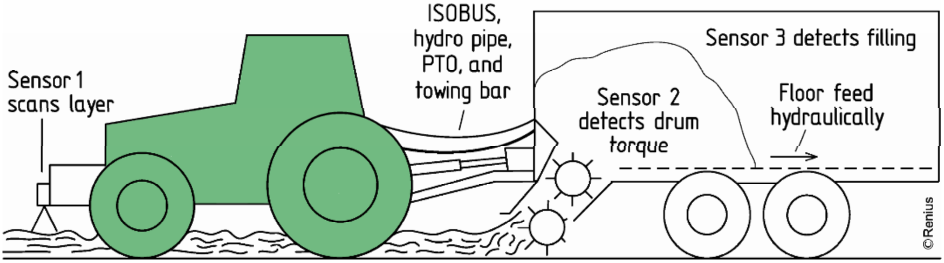
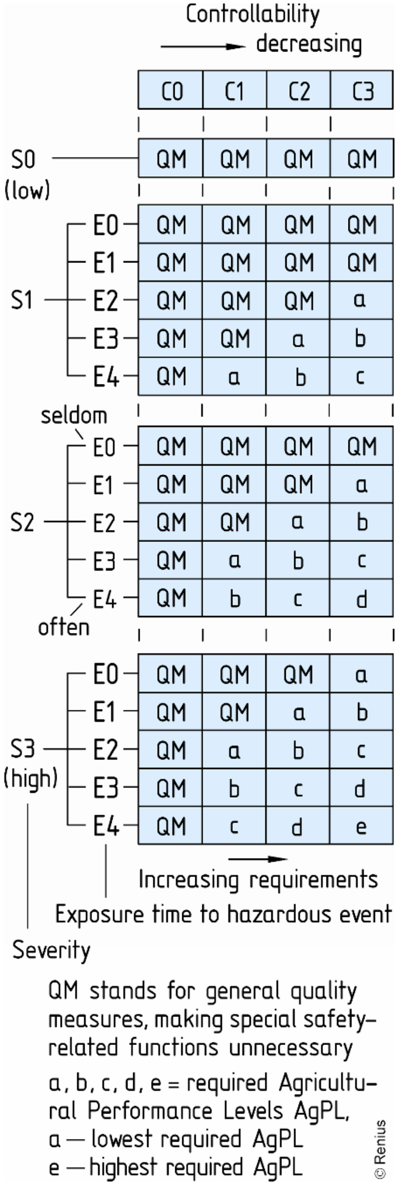


Fig. 7.5: Implement guided tractor control: Sensor 1 detects the layer cross-section, achieving constant throughput by variation of tractor speed at constant PTO speed, which needs a tractor with CVT. Sensor 2 detects drum torque, which is kept constant by adjusting the trailer floor feed via tractor remote hydraulics. Closed loop control via ISOBUS improves field performance and reduces risks of material flow blockage. Proposed by Pöttinger and John Deere

Table 7.5: Deriving *Agricultural Performance Levels* for a mechatronic system according to ISO 25119-2:2019.

Inputs: - Severity S
- Exposure time E
- Controllability C

DIN V 19250:1994 was cancelled


Controllability
decreasing
seldom
often
Increasing requirements
Exposure time to hazardous event
Severity

		C0	C1	C2	C3
S0 (low)		QM	QM	QM	QM
S1	E0	QM	QM	QM	QM
	E1	QM	QM	QM	QM
	E2	QM	QM	QM	a
	E3	QM	QM	a	b
	E4	QM	a	b	c
S2	E0	QM	QM	QM	QM
	E1	QM	QM	QM	a
	E2	QM	QM	a	b
	E3	QM	a	b	c
	E4	QM	b	c	d
S3 (high)	E0	QM	QM	QM	a
	E1	QM	QM	a	b
	E2	QM	a	b	c
	E3	QM	b	c	d
	E4	QM	c	d	e

QM stands for general quality measures, making special safety-related functions unnecessary

a, b, c, d, e = required Agricultural Performance Levels AgPL,
a — lowest required AgPL
e — highest required AgPL

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Specific safety requirements have to be considered in tractor design, especially if electronic systems are employed in automated systems (Table 7.4). As a general recommendation, the development of integrated safety systems should be part of the general development process as illustrated in the V-model (Fig. 6.61).

Basic principles of **risk assessment** of technical systems are standardized in EN ISO 14121:2008.

Parts 1 – 4 of ISO 25119 have been developed for tractor-implement and forestry systems, updated in 2019.

Part 1 defines: *The key objective is that the designers ensure that the safety-related parts of a control system produce outputs which achieve the risk reduction objectives of ISO 14121.*

Part 1 **defines the risk** as

a combination of the probability of occurrence of harm and the severity of that harm.

Part 2 contains a procedure to derive consequences for specific cases, which are addressed by input symbols C, S, and E, **Table 7.5**. The symbols have the following meanings:

- **C** *Controllability*: potential driver reaction to prevent a hazard, C0 means perfect control, C3 no chance of control.
- **S** *Severity* of the case: S0 means “no significant injuries, requires only first aid”, S3 means “survival uncertain”.
- **E** *Exposure time*: E0 means that the critical situation is expected “extremely seldomly”, E4 means “frequently”.

QM means in Table 7.9 that the quality measures of the basic design exclude hazards. Symbols a – e represent an *Agricultural Performance Level AgPL* which requires specific safety-related functions. To cover unknown relevant risks of implements, an AgPL one step higher is recommended by experts. Hard- and software development see ISO 25119, part 3.

7.3 Tractor-implement interface

In its first development phase, the agricultural tractor was mainly used for pulling implements and trailers and driving threshing machines. This has changed.

The first powertrains for implements only worked mechanically; however, modern tractors also offer other power transmissions by hydrostatics, up-coming electrics, and in some cases by pneumatics [7.1].

The most important interfaces in all countries are the three-point hitch and the rear PTO. Additional connecting elements for trailers are important in some regions such as Western Europe or developing regions such as Africa, China, or India.

Premium tractors offer more functions; a rear interface for a modern Western European tractor is shown in **Fig. 7.6**. Other markets favour drawbars to pull large implements with less priority for trailers.

Table 7.6 summarizes the most important interface components. The listed standards enable various combinations of tractors and implements worldwide even from different manufacturers and countries.

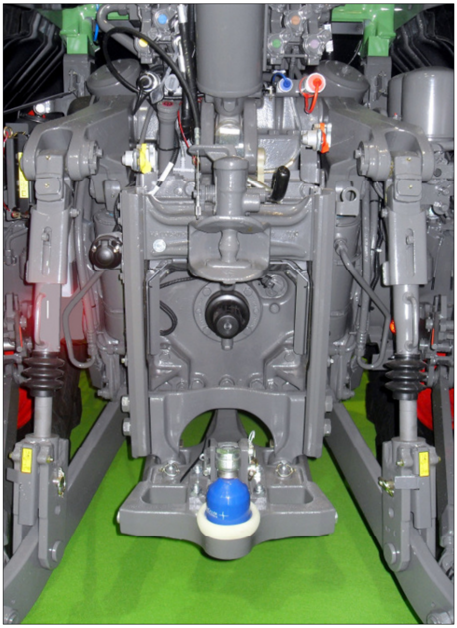

Fig. 7.6: Rear interface of a modern tractor with PTO, three-point hitch, clevis coupler (trailer, movable up and down), 80 mm ball coupler, sockets for hydraulic remote control, electric power, ISOBUS, etc. Photo: Author

Table 7.6: Survey of tractor-implement interfaces for power transfer. Electric connectors for 48 V and high voltage (AC, DC) are expected but not yet standardized

Mechanics	Hydrostatics	Electrics	Pneumatics
Rear elements: - 3-point hitch (ISO 730) - Hitch hook (ISO 6489-1) - Clevis 40 coupler (ISO 6489-2) - Drawbar (ISO 6489-3) - Piton fix (ISO 6489-4) - Clevis, non-swivelling (ISO 6489-5) - 80 mm ball (ISO 24347) - Rear PTO (ISO 500)	- Standard quick-action couplings (ISO 10448, ISO 5675, ISO 16028)	- 7-pole connector 12 V (ISO 1724)	- Connectors in general (ISO 6150)
- Front 3-point hitch and front PTO (ISO 8759)	- Power beyond couplings (ISO 17567)	- Integrated 12 V connector (ISO 11783)	- Connector for trailer brakes (truck technology)
	- Couplings for trailer brakes (ISO 5676)	- Single 48 V connector	- Connectors for tire inflation control
		- Single high voltage AC connector	- Connectors for pneumatic actuators (ISO 4414)
		- Single high voltage DC connector	

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The following two pictures demonstrate typical practical situations regarding the tractor interface. The cab of **Fig. 7.7** allows a certain access to rear interface elements**.** When the driver dismounts the tractor to connect remaining couplings, he (or she) can operate several interface functions using buttons on the fender. A typical configuration of a PTO drive line from tractor to implement with universal joint shaft is shown in **Fig. 7.8**.

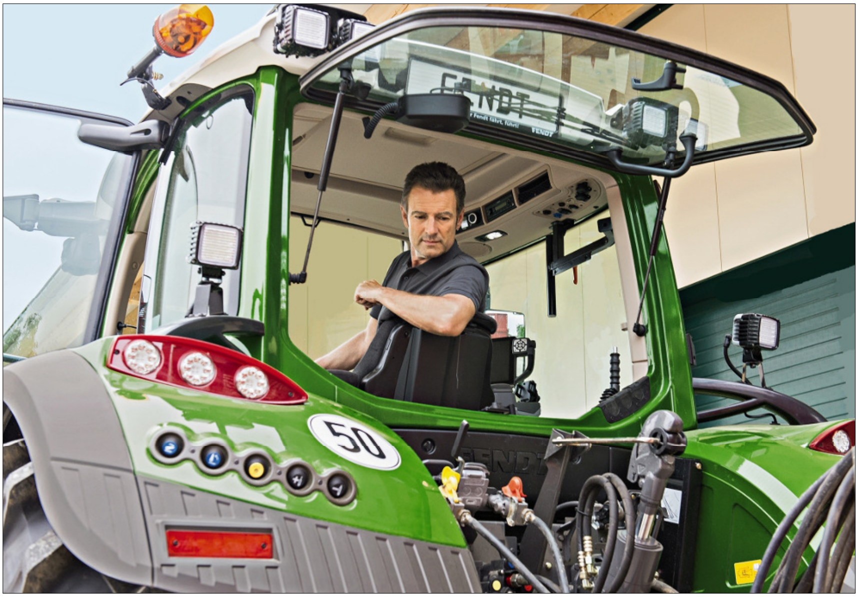


Fig. 7.7: Rear cab window open for access to elements of the rear interface. When the driver dismounts the tractor he (or she) can, for example, operate PTO, hitch, and hydraulic valves by-buttons on both fenders left and right, typical for premium tractors. Courtesy AGCO-Fendt

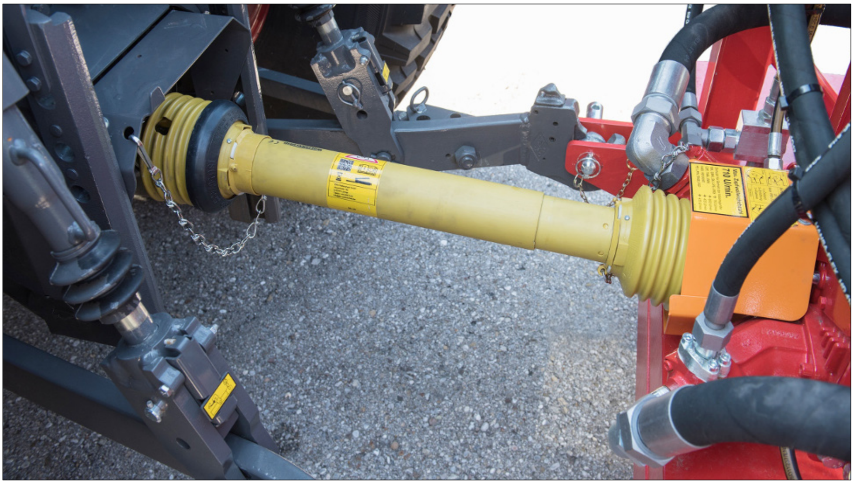


Fig. 7.8: Typical PTO drive shaft between tractor and implement. Courtesy AGCO-Fendt

Many types of mechanical connections are in use [7.2], the most important of which are specified by ISO, **Table 7.7**.

They support a worldwide compatibility between tractors and implements improving economics in agricultural mechanization.

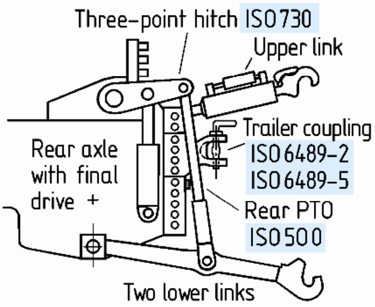
All these connectors should meet standardized relations to the PTO shaft position, mainly in length (for drawbar see Fig. 5.15). This is important as many implements or trailers need access to both, the PTO and a towing connection.

Table 7.7: Basic mechanical tractor-implement interface components and related ISO standards

Basic mechanical connections
(standard in many countries)

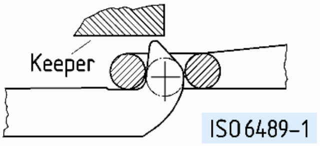
There is not enough space to install all standardized elements at the same time. However, extra flexibility is obtained by putting the trailer coupling in vertical rails, moving it on demand up or down for adequate PTO clearance and safety.

The length of the upper link of the three-point hitch can be adjusted manually or hydraulically, optionally also with integrated damping function for smooth implement transportation.

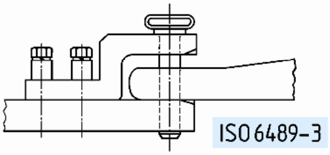


Special mechanical connections

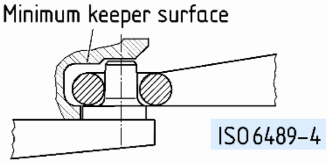
Hitch hooks can be moved up and down, linked to the three-point hitch mechanism. ISO 6489-1 specifies dimensions for vertical loads ≤ 30 kN.
Application for non-balanced trailers.



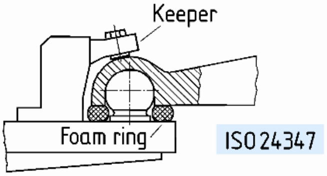
Drawbars are standardized in six categories for maximum nominal vertical loads of 5 to 30 kN. Their length may be adjustable and there are versions that swivel horizontally.
Application mainly for large tractors pulling large implements.



Piton fix is also a connection for non-balanced trailers, only used in a few markets. ISO 6489-4 contains dimensions for vertical loads ≤ 30 kN and specifies maximum keeper conture.



The 80 mm ball-type coupling for agriculture allows nominal vertical static loads of 13–33 kN (40 kN of first issue revised!).
"Short position", PTO type 1 and 2: 27 kN
"Short position", PTO type 3 and 4: 33 kN



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Universal joints and drive shafts play an important role for the powertrain from the tractor PTO shaft to PTO-driven implements [7.3]. Design and calculation fundamentals were presented with the support of GKN Walterscheid in [7.4]. PTO drive shafts are designed telescoping to deal with changing geometric combinations of tractor and implement without any constraints or damaging effects.

Simple universal joints produce velocity variations when tilted, **Table 7.8** (upper part). *They create torque vibrations with high peak torques; hence, reduce the service life of the whole PTO powertrain.*

Velocity variations can be compensated for by special configurations of the joints, **Table 7.8** (lower part). Wide-angle constant velocity universal joints are derived from the W-type of Table 7.8, **Table 7.9**.

Table 7.8: Basic configurations of PTO drive shafts between tractor and implement with single universal joints. Incorrect arrangements produce vibrations with oscillating extra torques which can damage the drive line. Related mathematic fundamentals are presented in [7.3] and [7.4]

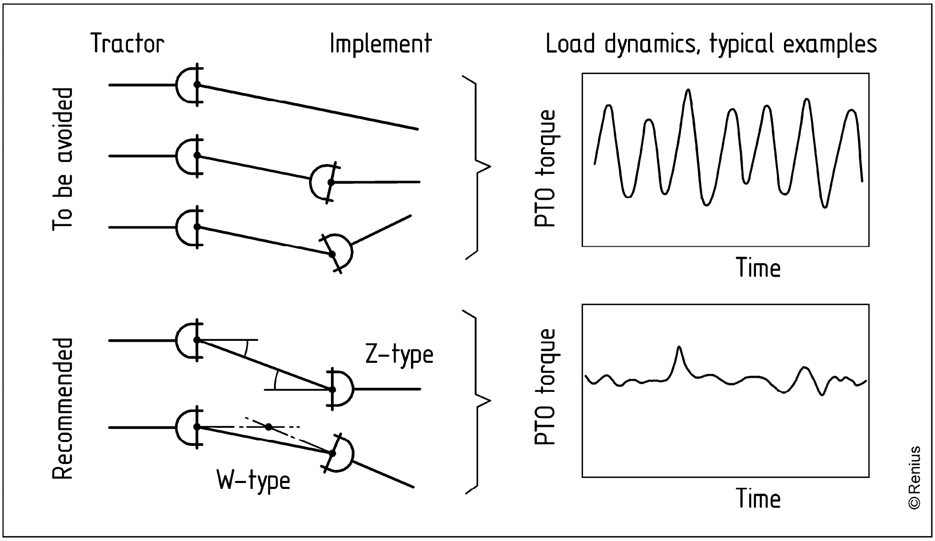
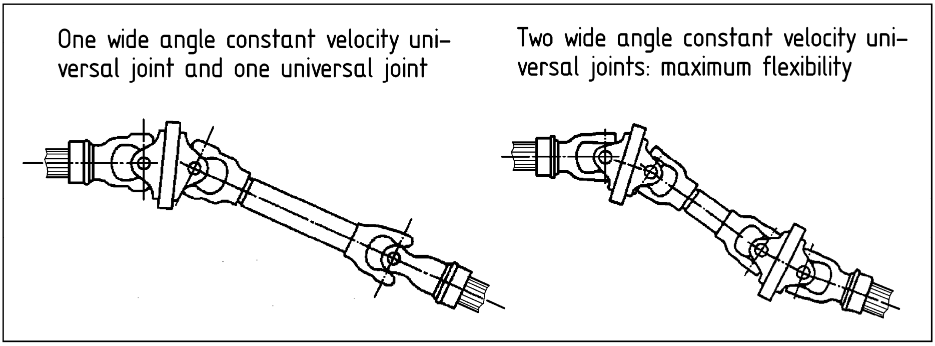


Table 7.9: Wide-angle constant velocity universal joints with internal W-type compensation allow high flexibility and low risks of misuse for PTO powertrains between tractor and implements


One wide angle constant velocity universal joint and one universal joint
Two wide angle constant velocity universal joints: maximum flexibility

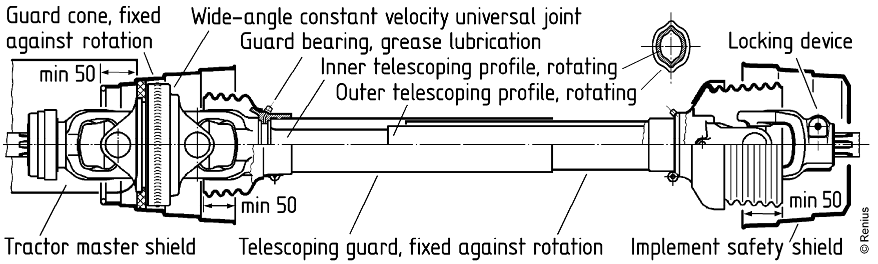

Guard cone, fixed against rotation
Wide-angle constant velocity universal joint
Guard bearing, grease lubrication
Inner telescoping profile, rotating
Outer telescoping profile, rotating
Locking device
min 50
min 50
min 50
Tractor master shield
Telescoping guard, fixed against rotation
Implement safety shield
© Renius

Fig. 7.9: Typical PTO drive shaft between tractor and implement. Popular configuration as shown in Table 7.9, left type. Attention should be paid to the safety elements protecting the rotating parts and to the telescopic shaft profiles as well, which need good lubrication to limit thrust forces. Safety elements are defined in ISO5673-1 and EN12965. Several sources

The left drive shaft configuration of Table 7.9 is very popular; hence, shown in more detail in **Fig. 7.9**. All PTO drive shafts between tractor and implement must be designed with telescoping elements in order to allow length variations in turns or during other implement movements.

Unguarded drive shafts are dangerous as human contact can create severe injuries. Safety elements were developed to cover the complete rotating PTO shaft system. ISO 5673-1 and EN 12965 contain detailed specifications which are used for safety approvals in many countries.

Thrust forces can be a problem, mainly under full load. ISO 5673-1 recommends, for example, a maximum value of ±12 kN for PTO type 1 and 2 and ≤ 115 kW shaft power (Table 5.8). This requires proper grease lubrication of the sliding profiles.

The thrust force F_{ax} can be calculated as follows:

$$F_{\mathrm{ax}} = \frac{T}{r} \cdot \frac{1}{c} \cdot \mu \qquad (7.1)$$

where

T = torque of PTO drive shaft
r = effective radius of circumferential force
μ = coefficient of friction (lubricated ≈ 0.1)
c = coefficient of profile geometry (< 1)

Hence, high thrust forces can result from a small effective radius (r), unfavorable geometry (c), and poor lubrication (μ).

A wide-angle constant velocity universal joint is shown in **Fig. 7.10** and **7.11**.

Its kinematics are based on two simple universal joints of the W-type (Table 7.8) with an ultra-short connecting shaft.

The W-type geometry is forced by the movable disc in a conducting chamber.

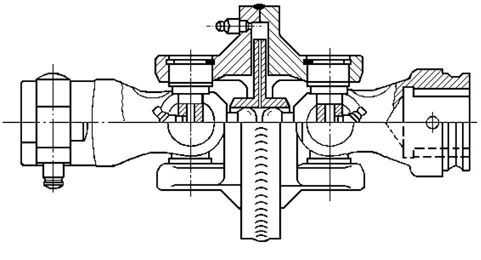


Fig. 7.10: Wide-angle constant velocity universal joint (W-type, Table 7.8).
From [7.4], modified

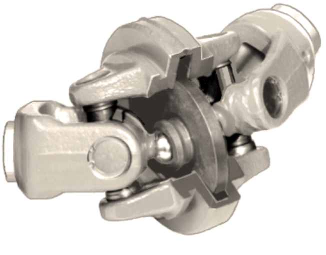


Fig. 7.11: Wide-angle constant velocity universal joint (W-type,Table 7.8).
Courtesy GKN-Walterscheid

7.4 Three-point hitch: Structure, standards, survey on controls

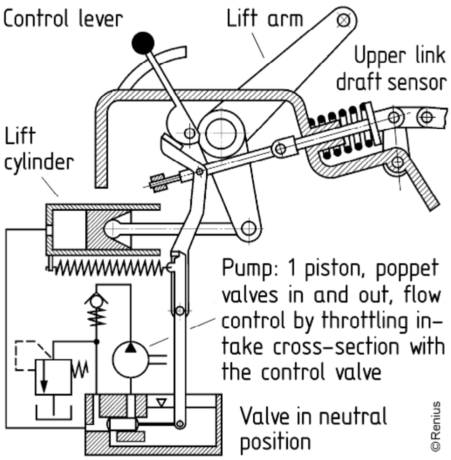

Control lever
Lift arm
Upper link draft sensor
Lift cylinder
Pump: 1 piston, poppet valves in and out, flow control by throttling intake cross-section with the control valve
Valve in neutral position
©Renius

Fig. 7.12: Early structure of the *Ferguson* system for implement draft control using an upper link sensor to convert compressive force to movement. Position of control lever represents the pull force target (after [7.8])

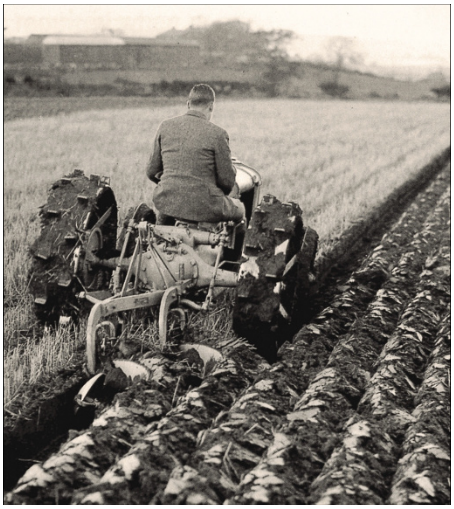


Fig. 7.13: Tractor of David Brown-Ferguson (20 hp) pulling a plow. This was the first commercial application of the *Ferguson* system. Two-furrow plows were the standard for such tractors at that time. Courtesy AGCO-Valtra

The Fordson F tractor (chapter 1.3) was ahead in many technical aspects at its time but did not satisfy all practical criteria regarding the connection of plows.

This motivated *Harry Ferguson* (1884–1960) to develop improved structures. He submitted his first patent as early as 1917; at least eight others followed [7.5]. Some of them were addressed in chapter 1.3.

Patent No. 226,033 [7.6] (filed in 1923) contains the connection of a plow with the tractor rear by a three-point linkage while patent No. 253,566 (filed in 1925) includes the well-known draft control [7.7]; hence, it can be called *Ferguson's* master patent.

Ferguson's idea was to carry and control a mounted plow automatically by the tractor with the following benefits:

- Lightweight tractors for low initial costs
- Weight transfer for better traction
- Improved quality of plowing
- Prevention of "plow sticking in the mud"

The proposed signal was either the implement force measured by springs or the torque of the final drive (for example, by the thrust force of a worm drive).

Ferguson proposed three types of actuators for closed loop draft control [7.7]:

- Mechanical
- Electrical
- Hydrostatic

In 1936 he improved his system by his British patent No. 470,069, **Fig. 7.12**, realized in the same year in series production of Ferguson-Brown tractors; hence, the first commercial application, **Fig. 7.13**.
The control lever was used for both, moving the plow up and down on the headland and setting the control target.

The *Ferguson* system is one of the most important applications of closed loop controls in agricultural mechanization. Closed loop controls became very important for many other applications as major pillar of precision agriculture and smart farming.

The modern three-point hitch has one central or two outside cylinders, **Fig. 7.14**.

The upper link, the lift rods, and the stabilizer are usually adjustable in length. Dimensions are standardized, **Table 7.10**.

The **nominal lifting force** at the lower hitch points should be *about equal to the tractor net weight for tractors using mounted implements.* ISO 789-2 [7.9] defines the nominal value for 90% relief valve setting.

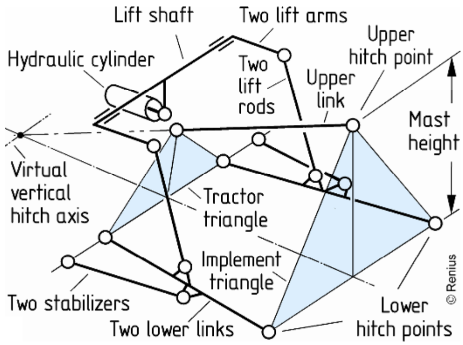
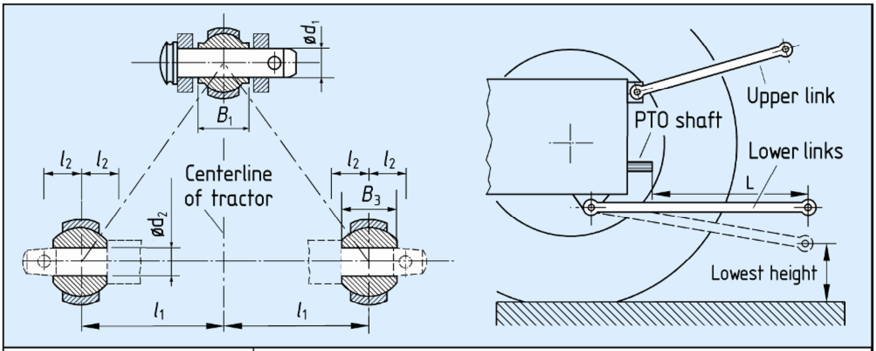


Fig. 7.14 (right): Linkage of three-point hitch

Table 7.10: Rear three-point tractor hitch: interface to implements as standardized in ISO 730



	Category							
	1N	1	2N	2	3N	3	4N	4
Nominal kW at PTO	≤35	≤48	30–92		60–185		110–350	
Upper hitch point	mm							
d_1 Dia. of pin hole	$19.3^{+0.2}_{0}$		$25.7^{+0.2}_{0}$		$32.0^{+0.25}_{0}$		$45.2^{+0.3}_{0}$	
B_1 Width of ball	$44.0^{0}_{-0.5}$		$51.0^{0}_{-0.5}$				$64^{0}_{-0.5}$	
Lower hitch points	mm							
d_2 Dia. of pin hole	$22.4^{+0.25}_{0}$		$28.7^{+0.3}_{0}$		$37.4^{+0.35}_{0}$		$51^{+0.5}_{0}$	
B_3 Width of ball	$35.0^{0}_{-0.5}$		$45.0^{0}_{-0.5}$				$57.5^{0}_{-0.5}$	
l_1 Distance from tractor centerline	218	359	364	435		505		612
l_2 Lateral movement	≥50	≥100 *)		≥125				
L Horiz. distance (35 Ø PTO shaft)	350^{+25}_{-50}	550^{+25}_{-50}	600^{+75}_{-90}		650^{+75}_{-90}		Not recommended	
L Horiz. distance (45 Ø PTO shaft)	Not recommended		600^{+125}_{-10}		650^{+125}_{-10}		750^{+125}_{-100}	
Lowest height of lower hitch points	200 max.		230 max.					

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*) If the tractor has a track width ≤1150 mm, this value may be reduced to 50 mm min.
Mast height is not standardized for the tractor triangle, but for the implement triangle

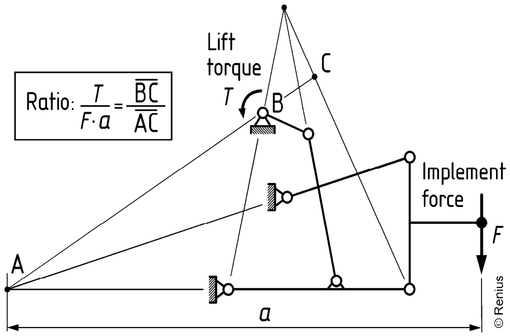
Ratio: $\frac{T}{F \cdot a} = \frac{\overline{BC}}{\overline{AC}}$
Lift torque
T
B
C
A
Implement force
F
a
© Renius

Fig. 7.15: Graphic solution of the three-point hitch ratio [7.11]. Mathematical models see [7.11–7.15]. Lifting capacity is defined in ISO 789-2 : 2018 [7.9]. Nominal pump pressure is 200-210 bar according to limits for implements by ISO 10448:1994 [7.16]

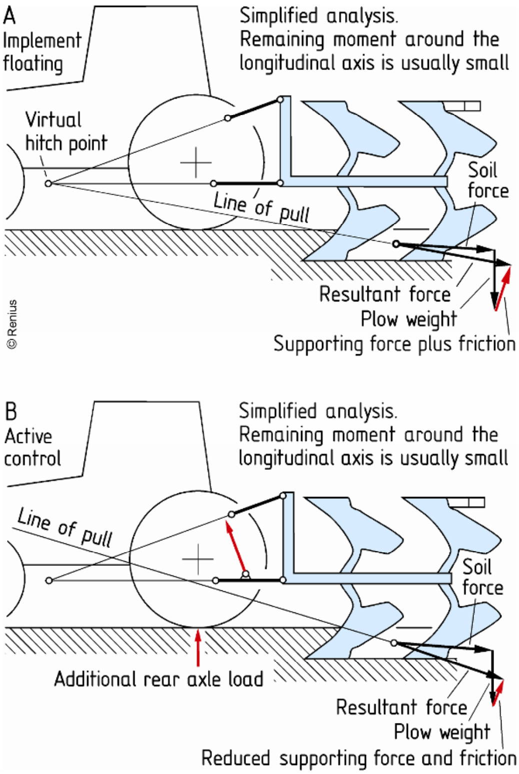
A
Implement floating
Simplified analysis.
Remaining moment around the longitudinal axis is usually small
Virtual hitch point
Line of pull
Soil force
Resultant force
Plow weight
Supporting force plus friction
© Renius
B
Active control
Simplified analysis.
Remaining moment around the longitudinal axis is usually small
Line of pull
Soil force
Additional rear axle load
Resultant force
Plow weight
Reduced supporting force and friction

Fig. 7.16: The two basic cases of connecting mounted implements such as plows with the three-point hitch system. A floating position, B implement draft control by the three-point hitch control valve

The **kinematics of the linkage** can be simplified in a two-dimensional (plane) diagram, **Fig. 7.15**. If a certain vertical force, for example implement weight *F*, is given, the lift torque can be derived from the distances BC and AC [7.11]. *Full load efficiencies (cylinder pressure to implement)* are about 80-85 % due to friction losses [7.12–7.14].

Three-point hitch controls can be classified roughly into two groups, A and B, **Fig. 7.16**.

The simple system **(A)** connects the implement in an up-down floating position only; no lifting forces during work. The implement is self-conducting in depth according to the equilibrium of all implement forces, mainly supporting forces. The line of draft crosses the virtual hitch point. As there are almost no vertical loads transferred to the tractor, there is no “Ferguson effect”; hence, almost no additional vertical tire loads for improved traction. However, a simple improvement can be achieved by a slight constant pre-pressure on the system, resulting in carrying a certain fraction of implement weight and hence improving traction.

A logical use of Ferguson’s invention needs a force sensor to keep the pull force constant via the control valve (Fig. 7.12). The implement is now mostly carried by the tractor with only low supporting implement forces, as shown in Fig. 7.16 **B**. This now improves traction and prevents sticking.

However, Ferguson’s principle still has some disadvantages:

1. If the field soil density increases in spots during tillage operations, the system lifts the implement, *reducing tillage quality*.
2. Simple mechanical draft control systems can be *unstable at high control amplification factors* [7.14, 7.17, 7.18].

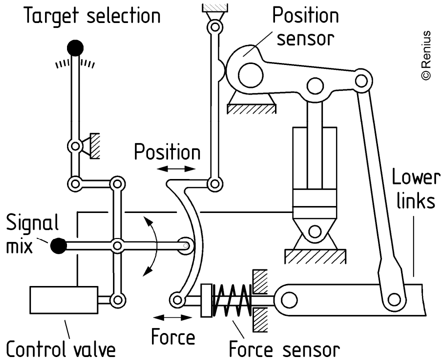

Fig. 7.17: "Mixed" control, compensating disadvantages of the simple draft control

An improved system, based on a patent of John Deere [7.19], measures the implement position by a second sensor and mixes this signal with the force signal, **Fig. 7.17**. The mix can be varied continuously between 100% force and 100% position control.

The mixing effect is a compromise: It smoothes the variations of implement working depth but, at the same time, reduces the sensitivity of the draft control. However, this characteristic leads to a welcomed *increased dynamic stability of the closed loop control* [7.14].

Table 7.11: Survey of sensors possible for force measurement [7.20]. The marked areas indicate practical applications

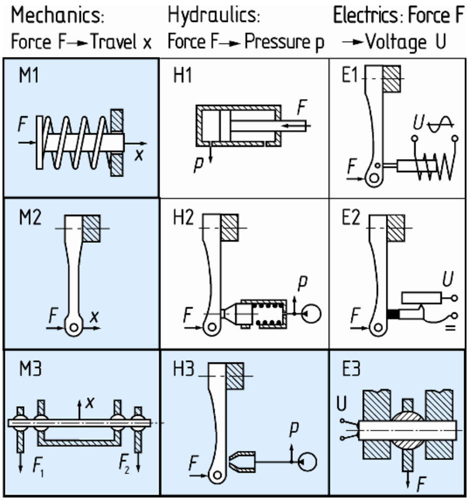
Mechanics: Force F→Travel x
Hydraulics: Force F→Pressure p
Electrics: Force F →Voltage U
M1
F
x
H1
F
p
E1
U
F
M2
F
x
H2
p
F
E2
U
F
=
M3
x
F1
F2
H3
p
F
E3
U
F

Table 7.11 shows a survey of *Hesse* [7.20] on force transducers (F force, x travel).

M1 represents the Ferguson sensor, M3 is a cheap lower link sensing (John Deere). E3 is part of the Bosch EHR and H3 was used for Rexroth-IH SHR [7.21]. The number of potential control types has increased along with progress in electronics, **Table 7.12**.

Table 7.12: Basic tractor hitch control strategies, among which the positions 1 to 7 dominate

Control strategy	Typical applications, comments
1. Free-floating hitch	Self-conducting implements, control valve in sinking position
2. Force control by upper link	Low-cost system for mounted implements, mainly plows
3. Force control by lower links	System for mounted and semi-mounted tillage implements
4. Position control	Mounted implements, e. g. for distribution of seeds, chemicals
5. Mix of force and position control	Systems 2, 3, 4 improved, mainly for tillage implements
6. Wheel slip control	Mainly all types of tillage implements with heavy pull
7. Hydraulic pressure control	Load transfer from implements or trailers improving traction
8. Controls with implement sensors	Implements with depth sensors
9. Controls using external cylinders	Semi-mounted tillage implements with depth wheel control
10. Transmission torque control	Tillage implements (disturbed by vehicle traction mechanics)

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7.5 Global design demands, loads, control systems, electronics

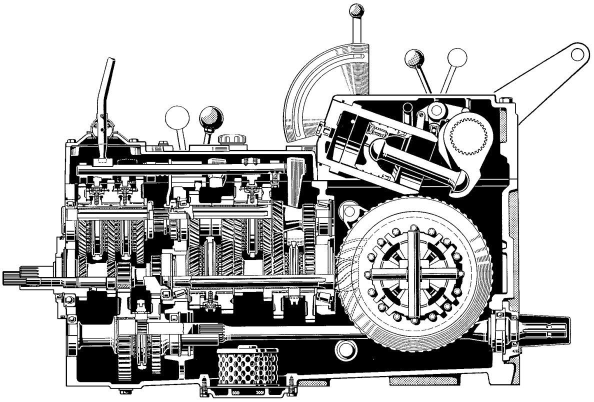

Fig. 7.18: Integrated hydraulic hitch cylinder of transaxle ZF T 300 (later T 3000) with 12/6 speeds, very popular in Europe for large tractors from 1966 to about 1994. Courtesy ZF

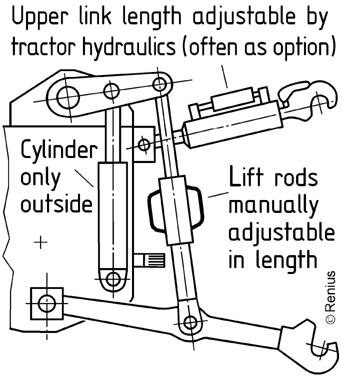
Upper link length adjustable by tractor hydraulics (often as option)
Cylinder only outside
+
Lift rods manually adjustable in length
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Fig. 7.19: Typical rear three-point hitch of a modern tractor. Hydraulically adjusted length of upper link is a popular option

From about 1960 to 1990, the typical tractor hitch had one large hydraulic cylinder, which was well integrated into the rear transmission housing, **Fig. 7.18**. A development by Deutz (KHD) achieved very high production volumes [7.22]. It can still be helpful for hitch developments for tractors of lower technology levels, **Table 7.13**.

Additional outside cylinders were used around 1970–1985 for large tractors. The typical present design for premium tractors is shown in **Fig. 7.19**. Maximum working pressures rose from about 70 bar in 1951 [7.8] to now 180–200 bar [7.22, 7.23] due to improved gear pumps with leakage compensation or modern axial piston pumps.

Table 7.13: Three-point hitch linkage functions and control, classified into five technology levels

	Hitch linkage							Force control				Position control			Mixed control		Slip control		Other controls		
Technology level	Cylinder integrated	Cylinder outside	Upwards forces only	Force also downw.	Upper link hydraul.	Lift rod length hydr.	Stabilizer hydraul.	Upper link, mech.	Lower links, mech.	Upper link, EHR	Lower links, EHR	Lift shaft, mech.	Lift shaft, EHR	Implement, electr.	Mechanically	Electron., EHR	Mechanically	Electron., EHR	Const. pressure	Pressure control	Active upper link
I	X	-	X	-	-	-	-	X	-	-	-	-	-	-	-	-	-	-	(X)	-	-
II	X	-	X	-	-	-	-	X	X	X*	-	X	-	-	-	-	-	(X)*	(X)	-	-
III	(X)	X	X	-	(X)	(X)	-	-	X	-	X	X	X	-	X	X	-	(X)	(X)	(X)	-
IV	-	X	X	(X)	(X)	(X)	(X)	-	-	-	X	-	X	(X)	-	X	-	X	-	X	(X)
V	-	X	X	(X)	X	(X)	(X)	-	-	-	X	-	X	(X)	-	X	-	X	-	X	X

(X) often as an option EHR: electro-hydraulic regulation (BOSCH) * Low-cost EHR

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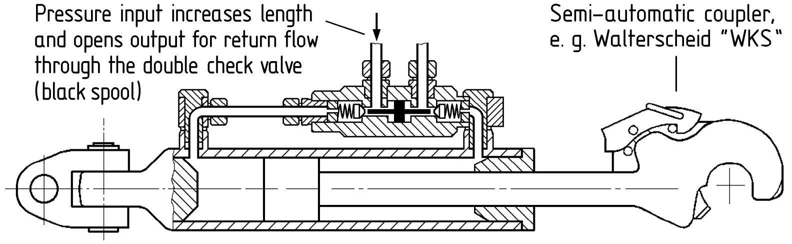
Pressure input increases length and opens output for return flow through the double check valve (black spool)
Semi-automatic coupler, e. g. Walterscheid "WKS"

Fig. 7.20: Hydraulically operated upper link for working with mounted implements, usually offered as an option. The double check valve closes with practically no leakage. Recent developments have integrated sensors for position feedback; for example, for keeping a fertilizer spreader horizontal or to improve plow control [7.24]. Courtesy GKN Walterscheid

Upper links of the three-point hitch can be ordered with hydraulic remote control, **Fig. 7.20**. The principle can also be applied to lift rods, even to the lateral stabilizer of the lower links. Most recent designs are offered with integrated position sensors.

Loads for dimensioning. Elements of three-point hitches have to bear high dynamic loads, mainly during driving on rough surfaces or over obstacles at high speeds with heavy mounted implements.

Lift rod forces can be used well to derive load forces of other elements such as cylinder loads (and pressures) or bending of lower links.

The measured load track of **Fig. 7.21** demonstrates the dynamics, in this case with a peak of 120 kN at an average static load of only about 35 kN. The frequency is about 1.8 Hz, relatively high because of the small distance of the ballast from the tractor rear and no front mass (see equation 6.4). **Fig. 7.22** confirms such peaks with forces up to 130 kN equal to 260 % of tractor net weight.

Considering unequal forces left and right, the maximum for the single lift rod for heavy mounted implements may peak at about 170 % and for top links at up to about 180 % of tractor net weight.

Dynamic tests should use rainflow load spectra; time portions only for control.

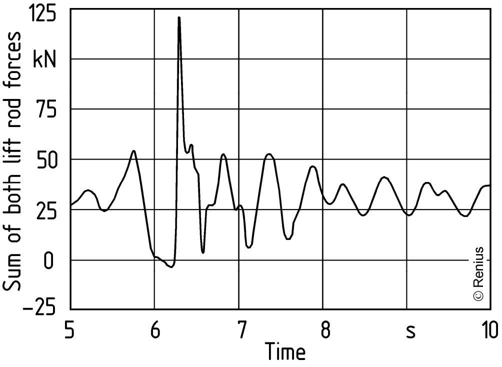

Fig. 7.21: Measured sum of both rear lift rod forces of standard tractor Fendt 509 C (70 kW). Net weight 5000 kg, front suspension blocked, rear ballast 1.7 t (center of gravity 0.425 m behind coupling triangle). Driving with 20 km/h over a ramp of 5.3 cm height and 2 m length [7.25]

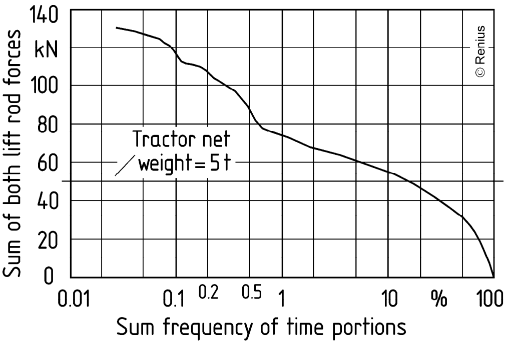

Fig. 7.22: Load spectrum of the sum of both lift rod forces of the rear three-point hitch for heavy conditions: driving 25 km/h on ISO 5008 "smooth track" with the tractor configuration as from Fig. 7.21 [7.25]

Performance of rear three-point hitches

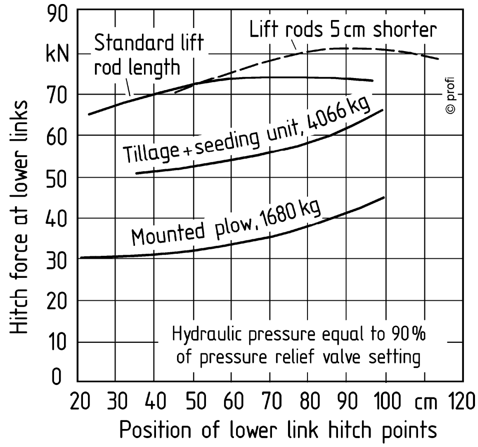

Fig. 7.23: Hitch forces at lower links of a Fendt 500 Vario tractor (120 kW EG 97/68), measured by DLG TestService GmbH, Groß Umstadt, Germany, on behalf of profi magazine, published in 2014. Data courtesy profi

Performance is usually measured in accordance with ISO 789-2 [7.9], which was harmonized in 2018 with OECD Code 2.

Both recommend measuring lift forces at the lower links with an adjustment of the linkage to achieve the power range and lower hitch point height as specified in ISO 730. Hydraulic pressure shall be equivalent to 90% of the relief valve setting.

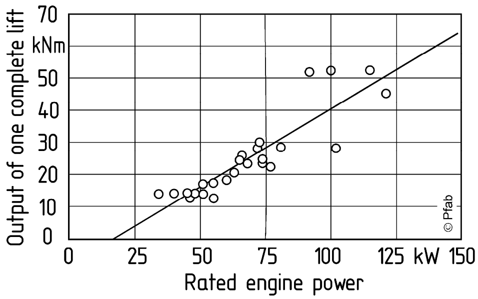

Fig. 7.25: Working capacity (output energy of one complete lift) of the rear hitch versus rated engine power. Values from OECD tests 1983–93 for 90% of relief valve pressure [7.14]

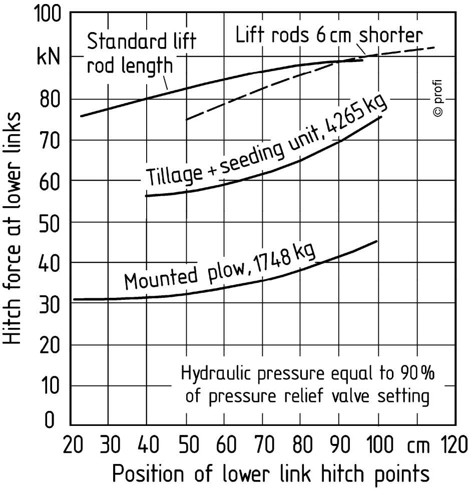

Fig. 7.24: Hitch forces at lower links of a Deutz-Fahr 6185 RCshift (133 kW ECE R 120) measured by DLG TestService GmbH, Groß Umstadt, Germany, on behalf of profi magazine, published in 2017. Data courtesy profi

Practical hitch forces are mainly important for heavy mounted implements.

Combinations of PTO-driven tillage implements with seeding machines represent the highest weights to be handled. Weights of mounted plows are usually much lower.

Measured performance curves are reproduced in **Fig. 7.23** and **7.24** and compared with demands of examples of the two mentioned implement groups.

The left diagram is from measurements of a four-cylinder standard tractor while the right one is from a standard tractor with a six-cylinder engine with a larger wheel base which can balance larger lift forces.

Output of one lift W was modelled by *Pfab* versus rated engine power P [7.14]:

$$W\,[\mathrm{kNm}] = 0.48 \cdot P\,[\mathrm{kW}] - 8.2 \qquad (7.2)$$

This equation can still be used as a guideline, **Fig. 7.25**, if heavy mounted implements are used (power acc. EG 97/68, pressure at 90% relief valve setting).

Control systems of the rear three-point hitch can be classified into four basic types:

1. *Mechanical two-point* valve control, *constant* displacement pump
2. *Mechanical throttling* valve control, *constant* displacement pump
3. *Mechanical proportional* valve control, *variable* displacement piston pump
4. *Electro-hydraulic proportional* valve control, *variable* displacement piston pump

Initial costs increase from 1 to 4. Types 1 and 2 are typical for low technology levels; 4 represents high levels IV and V.

Even simple low-cost control structures of class 1 or 2 require excellent expertise in design and production of hydraulics; see also the large number of patents.

Fig. 7.26 shows a type 1 control of the aforementioned Deutz hitch [7.22]. Pump flow is almost unloaded in the neutral position (valve 2 open). The check valve prevents leakage under load to keep an implement in an upper position for a longer time, a feature typical of all systems. When valve 1 of Fig. 7.26 is in its position "higher", pressure in the control line closes valve 2 and the pump working flow raises the hitch. When valve 1 is in the position "lower", return flow passes this valve directly via the flow control valve. There exist many patents describing circuits like Fig. 7.26, see survey in [7.14] (status of 1994, but still of interest).

Valve 2 shifts in a bistable way, reducing the loading time of the pump, hence works discontinuously.

Research by *Hesse* at FAL Braunschweig on electro-hydraulic hitch controls [7.26 7.27] paved the way for their commercial introduction, **Fig. 7.27**. In 1978, Bosch delivered the ever first EHR system for a MF 4WD tractor in the USA. The force sensor initially worked indirectly with inductive travel sensors; later on with magnetostriction. [7.28 7.29]. Main benefits are listed in **Table 7.14**, see also [7.30].

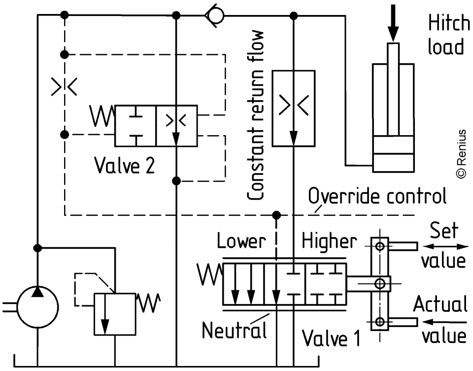

Fig. 7.26: Hydromechanical control of a Deutz rear hitch, presented in 1968 by *Koenig* [7.22]

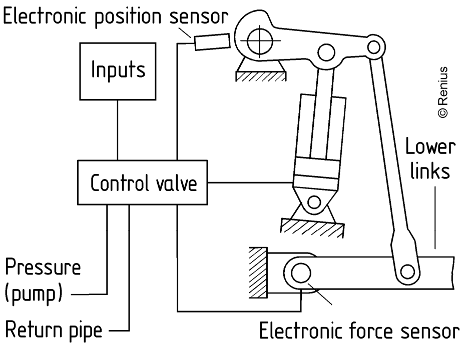

Fig. 7.27: Electro-hydraulic rear hitch control with electronic force and position sensors

Table 7.14: Benefits of electro-hydraulic three-point hitch control systems

- No complicated signal linkage
- Improved control dynamics through higher system stiffness, important for draft control
- Improved control quality by digital data processing and hence more flexibility in control strategies
- Improved flexibility in mixing signals
- Improved component flexibility
- No manual adjustments necessary
- Potential for driver information
- Potential for integration with tractor drive line and tractor-implement management systems (ISOBUS)
- Potential for diagnostics

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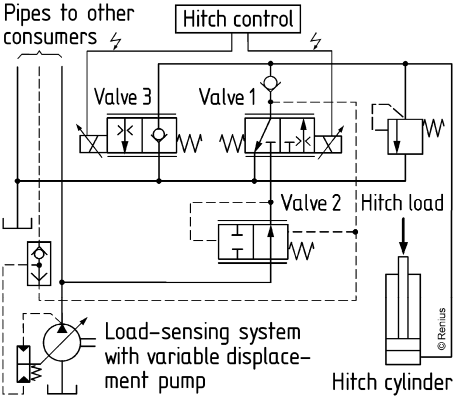

Fig. 7.28: Bosch electro-hydraulic hitch control for load-sensing circuits [7.29]

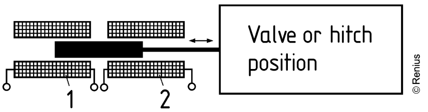

Fig. 7.29: Inductive position sensor

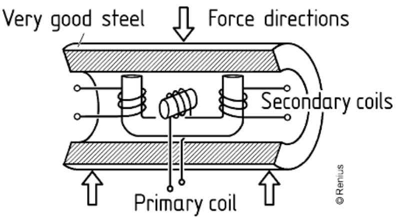

Fig. 7.30: Bosch force sensor. Drawing by the author based on material from Bosch

Table 7.15: Recommendations for high-quality hitch controls of premium tractors

- High stiffness of all loaded elements including oil volume
- Proportional valves with internal position control instead of shifting valves
- Valves with adequate dynamics
- Main valve with optimized dead zone
- Optimized amplification factor of the control unit, mainly for draft control
- Digital data processing, ISOBUS connection and access for diagnostics
- Draft controller able to shift PD ↔ PI
- Low energy consumption, low friction

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Electro-hydraulic control systems have been simplified and improved by the use of proportional valves, **Fig. 7.28** [7.29].

The load-sensing principle provides additional benefits as the pump always generates only the flow governed by the open section of valve 1 and *independently of hitch load and pump drive speed.* This improves control quality and saves energy [7.31]. Function details of *load-sensing* will be explained later on.

When an electric signal from the hitch control unit arrives at valve 1, it opens the flow to the cylinder according to the amount of the signal while valve 2 assists by keeping the pressure drop of 1 constant (details will be explained later on).

For lowering the hitch, valve 3 opens the return flow. Load influence on the hitch downward speed is corrected by overriding with the position control system.

Modern electro-hydraulic proportional valves work with internal feedback of the valve spool position, **Fig. 7.29**. Both coils of the sensor are supplied by high frequency AC (for example 5 kHz). A move of the ferromagnetic rod (black) to the right decreases the inductive resistance in coil 1 but increases it in coil 2. Both coils are part of a half bridge circuit.

Fig. 7.30 shows the principle of the present Bosch force sensor. The primary coil is supplied by high frequency AC, creating a magnetic field. Its transfer to the secondary coils is influenced by the shear stress of the steel pipe. This enables a proportional output signal with low hysteresis.

Table 7.15 offers recommendations for the design of updated hitch control systems.

The conflict between high sensitivity and dynamic stability requires proper design and control parameters, particularly for draft controls. Modern simulation tools and hardware-in-the-loop tests (chapter 8) can help to find acceptable results.

7.6 Front three-point hitch and front PTO

Front implements are very popular in some regions for tractors of technology levels III–V, in particular in Europe, **Fig. 7.31**, **7.32**.

The **front PTO** and its speed shall conform to type 1, 2, 3, or 4 of the rear PTO (ISO 500).

Special requirements are listed in ISO 8759. The direction of PTO rotation shall be *anticlockwise*, viewed from ahead of the tractor to the front part of the tractor. This simplifies the transmission between engine and PTO as it allows transmitting power with only one pair of gear wheels (engines run clockwise when viewed from the front). However, this means that rear implements cannot be directly attached to a front hitch. Some companies offer both directions on request. The direction of rotation and rotational speed shall be marked. Safety requirements and clearance zones around the front PTO should be designed similarly to those of the rear PTO.

The **front PTO** and **front hitch** are standardized in ISO 8759, 1–4. These standards are currently still under development. All four PTO types 1–4 are addressed, general specifications and locations listed in ISO 8759-3.

The most popular nominal rotational speed of front PTO is at present 1000 rpm, for small tractors alternatively also 540 rpm.

Front hitch forces should be about half of the rear hitch forces, mostly as an option, also with position control in some cases. The three-point hitch coupling of Fig. 7.31 and 7.32 was initially a development of the German company Weiste. It became very popular and was standardized in ISO 11001-2 [7.32].

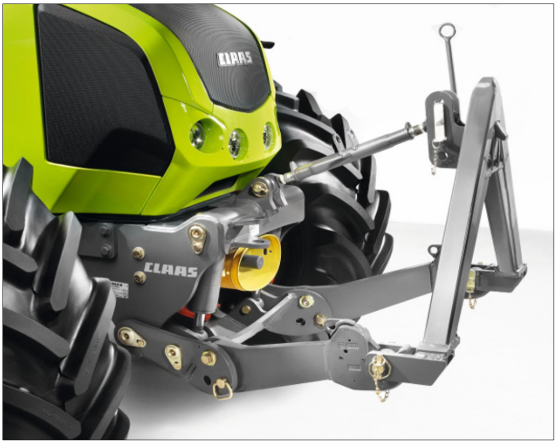

Fig. 7.31: Typical front interface of a modern tractor offering front PTO and front hitch with the male part of the “Weiste triangle”. Hydraulic and electric couplings are not visible: The lower links are usually rigidly connected together to offer high lateral stability for adequate implement guidance (by pushing). Courtesy CLAAS

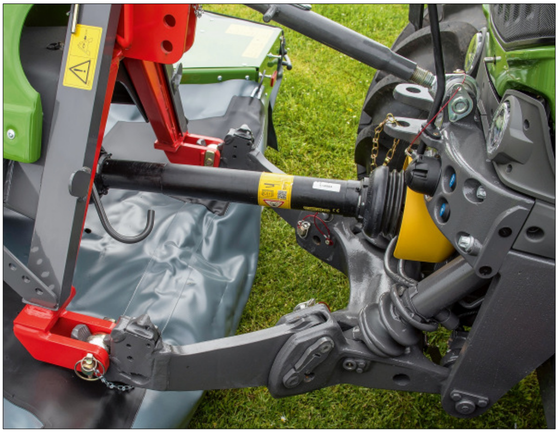

Fig. 7.32: Typical connection of a front implement using front hitch and front PTO, popular in Europe for forage cutting and light tillage implements. The “Weiste triangle” enables connection without leaving the cab but the PTO is still coupled manually. Courtesy AGCO-Fendt

7.7 Fundamentals of fluid power systems for tractors

7.7.1 Introduction

In his patents, *Harry Ferguson* detailed three options for his system: to realize it mechanically, electrically, or hydraulically.

Various designs did compete, even those with pneumatics (for the UNIMOG). But hydraulics were clearly able to win this race, due to their advantages, **Table 7.16**. Nevertheless, power transfer to implements gets competition from electrics, mainly for reducing energy losses of the power transformation. However, electrics will probably remain more expensive.

Developments in the automobile sector are going to reduce the costs also favouring combinations of electrics with hydraulics.

Table 7.16: Application of hydraulics to agricultural tractors, its advantages and disadvantages

Advantages of hydraulics

- Large variety of components available, high flexibility of package
- Most economic solution for linear actuators, mainly for high forces
- High dynamics because of very high power density
- Easy power transformation from rotation to translation
- Easy to build up closed loop controls with continuous action, hence good ability for automation
- Easy and cheap overload protection by pressure relief valves
- Oil transports heat and lubricates

Disadvantages of hydraulics

- Efficiencies lower than those of mechanical systems; in many cases also lower than those of electric systems
- Leakage results in a small reduction of output speed, requires control for perfect synchronisations
- Operational behavior is influenced by viscosity, hence by temperature
- Potential leakage needs very proper system design
- Often cooler needed
- Filters are necessary, their replacement requires maintenance
- Pumps and motors generate noise

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Table 7.17: Typical maximum rated pressures of tractor hydraulics

- Circuits for control, charging, filtering, and lubrication	10–20 bar
- Control variable Δp of analog load-sensing systems	20–30 bar
- Power steering, power brakes	100–205 bar
- Tractor hitch	205 bar
- Hydraulic pressure for tractor implements (ISO 10448)	205 bar
- Heavy duty swash plate pumps (power split CVTs)	420 bar
- Heavy duty bent axis units (power split CVTs)	450 bar

$10\,\text{bar} = 1\,\text{MPa} = 10^6\,\text{Pa} = 10^6\,\text{N/m}^2$

$1\,\text{psi} = 0.0689476\,\text{bar} \approx 0.069\,\text{bar}$

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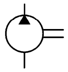
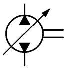
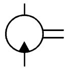
There are several pressure levels within tractor hydraulics, **Table 7.17**, with a maximum pressure for implements of “only” 205 bar. This value is fixed by ISO 10448 [7.16] to make sure that all implements worldwide are operated safely if connected via couplings with tractor hydraulics.

CVTs use about double that value, but their high pressure circuits are not directly combined with the circuit of the 205 bar consumers. Modern gear pumps and medium pressure swash plate pumps could offer slightly more than 205 bar but this is not used because of the mentioned reasons.

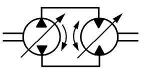
Hydraulic circuits can be explained well with standardized symbols, **Table 7.18**.

Table 7.18: Symbols for hydraulic elements as standardized in ISO 1219-1:2012 [7.33] (extract)

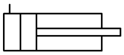
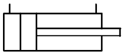
Pumps and motors

	Hydraulic pump with fixed displacement and one flow direction; for example, a gear pump
	Hydraulic pump with variable displacement and two flow directions; for example, a swash plate pump
	Hydraulic motor with fixed displacement and one flow direction; for example, a gear motor

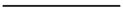
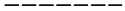
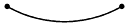
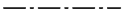
Hydrostatic transmission, HST

	Compact HST: Variable pump and variable motor with two output directions of rotation and variable output speed, e. g. for CVTs

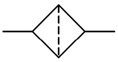
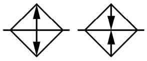
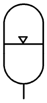
Hydraulic cylinder

	Single acting cylinder; for example, for lifting the rear hitch
	Double acting cylinder; for example, for front end loader tools

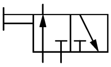
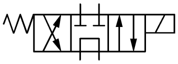
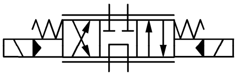
Connections and system border lines

	Pipe for energy transfer
	Control pipe
	Hose line
	System borderline for integrated components within a system

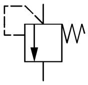
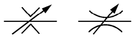
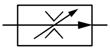
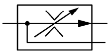
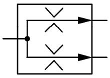
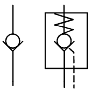
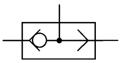
Circuit accessories

	Oil tank
	Filter
	Cooler, heater
	Accumulator, storing/delivering hydrostatic energy by loading/unloading a separated gas volume

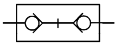
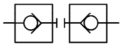
Directional control valves

	Directional valve 3/2: Three ports and two distinct positions, muscular control
	Directional valve 4/3: Four ports and three distinct positions, electric actuation against spring
	Directional valve 4/3 with intermediate positions and electro-hydraulic actuation, typical proportional valve

Other valves

		Pressure relief valve, limits pressure at input port by spring force, opens when input pressure force exceeds adjusted spring force. Internal control
	Flow control valves	Adjustable restriction valves, left turbulent, right laminar (capillary)
		Flow control valve: output flow constant and independent of pressure level, needs relief valve
		Flow control valve: output flow constant and independent of pressure. Bypass flow can be used for additional consumers
		Flow divider valve: Two output flows in a fixed ratio, independent of pressures
	Check valves	Left: Non return or check valve Right: Valve can be opened by control pressure, often applied to open a return flow of a cylinder
		Shuttle valve (OR function) in which the (left or right) inlet with the higher pressure is automatically connected to the (vertical) outlet

Quick-action couplings

	Quick-action coupling with two non-return valves, coupled
	Quick-action coupling, uncoupled

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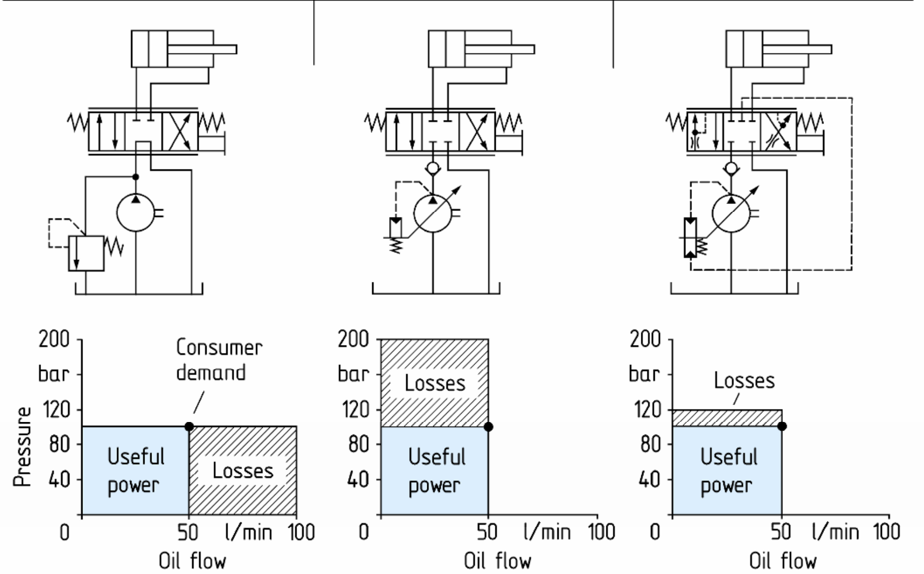
Tractor working hydraulics are usually realized in three typical circuits, **Table 7.19**.

Closed-center systems have closed directional valves in their neutral position. The most important example of a constant pressure system was that of John Deere, presented in 1960 [7.34] and produced for more than three decades. In 1973, Allis-Chalmers introduced the first load sensing system for tractors [7.35] recognizing already its benefits as listed in Table 7.19. Open-center systems are typical for tractors of the lower technology levels I–III and lower power levels while the more expensive closed-center systems are applied for premium tractors above about 75 kW.

Table 7.19: Basic circuit principles for tractor hydraulics and their typical properties. Comparison of typical losses was already well explained in 1973 by Allis-Chalmers engineers [7.35]

	Open-center systems Constant displacement pump(s), usually gear pump(s)	Closed-center systems with variable displacement pumps	
		Constant pressure system	Load-sensing system



	Open-center systems	Constant pressure system	Load-sensing system
Advantages	- Simple - Low cost - Many pump suppliers - Easy to maintain - Easy to repair	- More than one consumer operated independently - Disturbing factors "engine rpm" and "load pressure" are compensated within maximum pump flow capacity	
		- Quick response dynamics - Possibility of energy storage and recuperation	- Low throttling losses - Low heat generation
Disadvantages	- High losses - Only one consumer per pump independent (two with flow divider) - No infinitely variable closed loop controls	- High throttling losses, if consumer load pressure < system pressure - System permanently under pressure (leakage)	- The popular analog systems have poor dynamics - Throttling losses, if simultaneously supplied consumers vary in load pressure

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7.7.2 Basic circuits

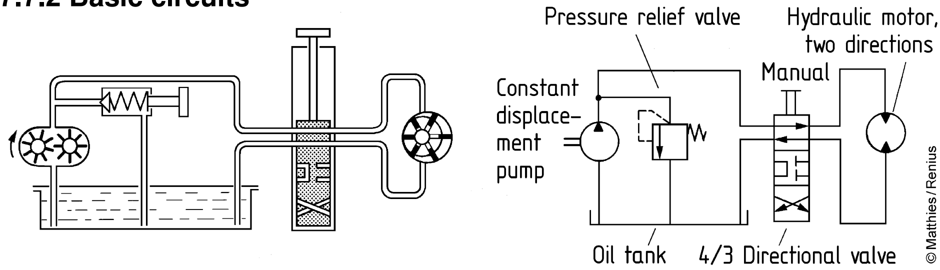

Fig. 7.33: Example of a simple open-center system. At left, drawing of a hydraulic transmission for two output directions. The central (neutral) valve position blocks the motor at zero speed and unburdens the pump via the U-turn. At right, the circuit with ISO 1219 symbols (Table 7.18)

The **open-center system** of **Fig. 7.33** shows a simple circuit schematically (left) and with ISO 1219 symbols (right).

Open-center systems were the first generation of applying hydraulics to tractor hitches, **Fig. 7.34**. Lifting is done by oil input under pressure to the cylinder, lowering occurs by the cylinder load along with an unloaded pump flow (U-turn).

With the introduction of power steering, a constant flow of about 8–12 l/min was needed, often realized by a flow control valve with the remaining bypass flow to the tank or preferably to the working hydraulics as shown in **Fig. 7.35** [7.36].

The first pump A supplies the working hydraulics (hitch, front end loader, and remote consumers) while the second pump B delivers the flow for the power steering. The pump is oversized delivering 10 l/min nominal flow even at engine idling.

Hence, there is a high flow surplus from pump B at higher (usual) engine speeds. This is now used for the working hydraulics. If the 3/2 valve is shifted to the left position, max. 60l/min are available, which is, for example, a welcomed level for high lifting speeds of a front end loader.

Open-center circuits have still potential, mainly regarding energy consumption, for example presented by an experimental digital multi-pressure circuit in [7.37].

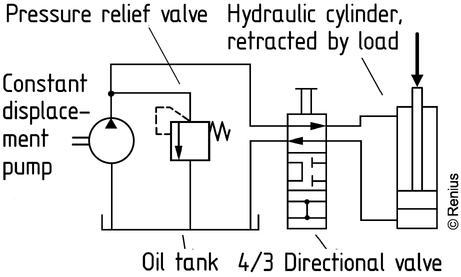
Pressure relief valve
Hydraulic cylinder, retracted by load
Constant displacement pump
Oil tank
4/3 Directional valve
© Renius

Fig. 7.34: Basic open-center circuit for hitches, also called "nudging type control"

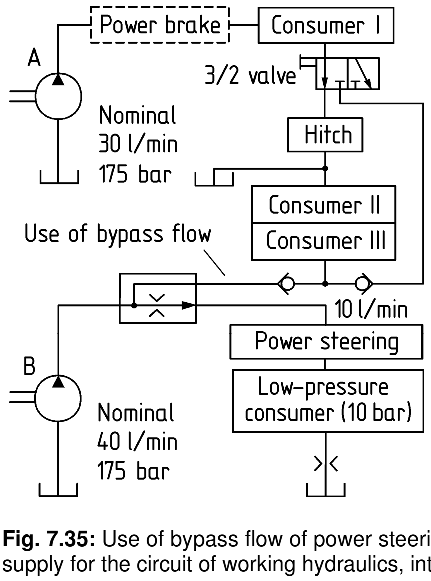

Fig. 7.35: Use of bypass flow of power steering supply for the circuit of working hydraulics, introduced by Fendt in 1980 [7.36]

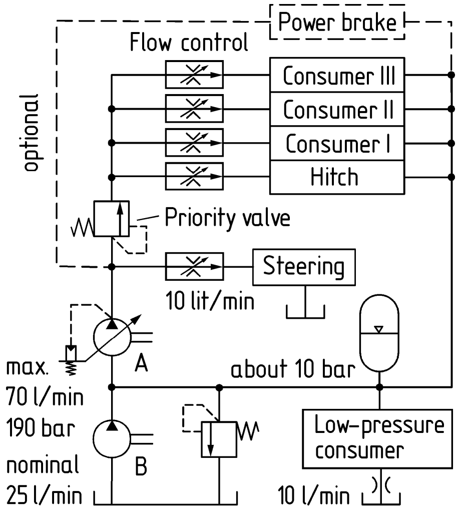

Fig. 7.36: Constant pressure circuit as introduced by John Deere in 1960 for the new tractors 3010 (60 PTO hp) and 4010 (84 PTO hp) [7.34]. Status 1979. After *Garbers* and *Harms* [7.36]

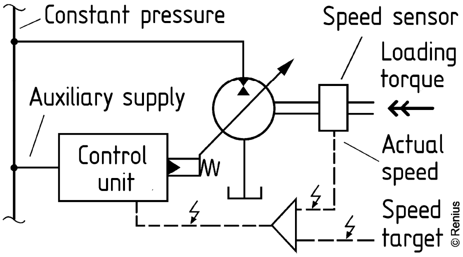

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Fig. 7.37: Principle of driving a hydraulic motor at a constant pressure circuit without throttling losses, known as “secondary control” [7.38]

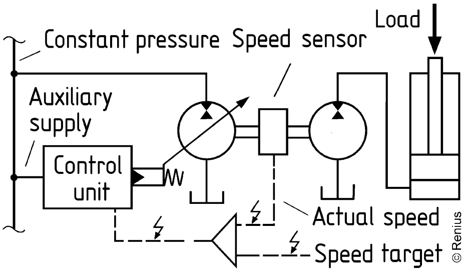

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Fig. 7.38: Principle of driving a hydraulic cylinder at a constant pressure net without throttling losses [7.38]

A **closed-center constant pressure hydraulic circuit** with variable radial piston pump was introduced by John Deere in 1960 for the new tractors 3010 and 4010 [7.34], **Fig. 7.36**. This system was produced in high volumes for more than 30 years. The output of pump A (partly charged by pump B) is controlled by its working pressure, automatically delivering the demanded flow. The priority valve safeguards the functions of power steering and power brake. All consumers can be operated independently within the limits of the maximum pump displacement.

Constant pressure systems became popular for a certain time. Ford presented, for example, a system with a constant displacement pump, charging an accumulator via a pressure-controlled on-off valve.

The main disadvantage is the energy losses of the flow control valves. They can be prevented by consumers with adjustable displacement which is, however, not applied at present to tractors. The principle is based on the balance between loading torque T, pressure difference Δp, and displacement of the hydraulic motor V (see also CVTs in chapter 5). If we neglect mechanical losses, the balance is:

$$T = \frac{\Delta p \cdot V}{2\pi} \tag{7.3}$$

Formulated with practical units:

$$T\,[\mathrm{Nm}] = \frac{\Delta p\,[\mathrm{bar}] \cdot V\,[\mathrm{cm}^3]}{20\pi} \tag{7.4}$$

In reality, mechanical efficiency must be introduced; guideline (full load): 90-94%.

Fig. 7.37 shows an application as used in aircraft hydraulics [7.38]. The balance is realized by regulating the output speed of the hydraulic motor by its displacement V. Operating a cylinder is also possible but more complicated, **Fig. 7.38**. Hydraulic transformers have been proposed for constant pressure circuits [7.39], but such circuits are not used at present for tractors.

Load-sensing systems with a constant displacement pump can have advantages over systems with flow controllers. If we compare **Fig. 7.39** [7.40] with Fig. 7.35, the flow to the consumer (often power steering) is now not constant, which is the main functional difference.

The first valve keeps the pressure Δp at the directional valve constant, which results in a flow being proportional to its opening section. This can be explained by the mathematical model of a turbulent flow Q through a sharp-edge orifice:

$$Q = \alpha \cdot A \cdot \sqrt{\frac{2\Delta p}{\rho}} \qquad (7.5)$$

The orifice flow coefficient α is nearly constant (often ≈ 0.7 for turbulent flow). The density ρ and the pressure drop Δp are also constant, which means that the flow rate Q is only a function of the valve opening section A. This offers not only proportional flow characteristics of the directionnal valve but also potential larger by-pass flows than with the system of Fig. 7.35.

Load-sensing systems with a variable displacement pump (Table 7.19) *realize the control of* Δp *directly by the pump*, **Fig. 7.40**. The pressures beyond valves 1 and 2 are directed to the shuttle valve 3; the higher pressure is then directed to the Δp valve of the pump which keeps Δp constant. This is possible with low throttling losses. If there are further consumers with lower load pressures (e. g. load II in Fig. 7.40), their Δp is not controlled by the pump but by the pressure compensator (5).

As each directional valve has such a pressure compensator, all the consumers can be activated independently within the limits of the variable displacement pump. The pressure compensators 4 and 5 are usually located upstream of the directional valve but a down-stream position beyond the directional valve is also possible.

In both systems, the two following important factors cannot disturb the control:

- *Engine rpm*
- *Load pressure(s)*

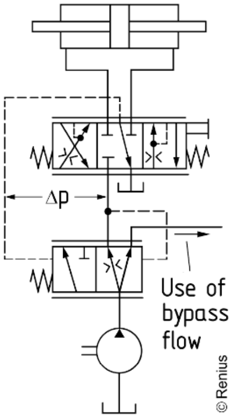


Fig. 7.39: Load-sensing circuit with a constant displacement pump, often applied for tractor power steering [7.40]

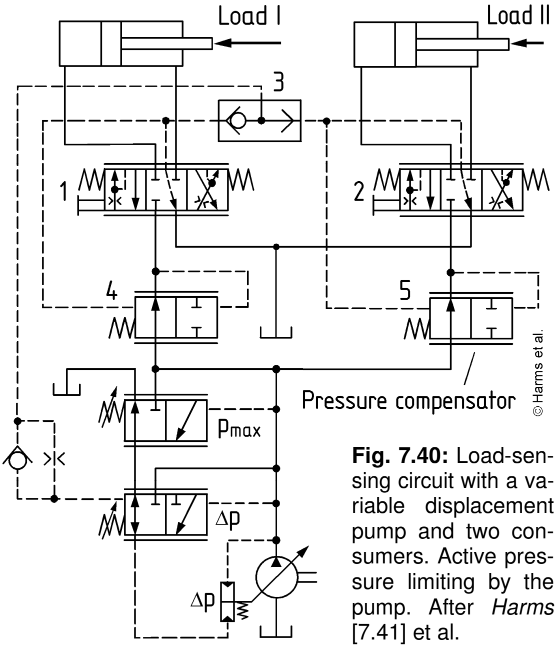


Fig. 7.40: Load-sensing circuit with a variable displacement pump and two consumers. Active pressure limiting by the pump. After *Harms* [7.41] et al.

7.7.3 Complete tractor circuits

Open-center circuits (Fig. 7.34 and 7.35) are still of interest for the lower technology levels. From 1960 to 1992, constant pressure hydraulics were the standard circuit principle of large John Deere tractors, but they are not applied to tractors at present (2019).

Hydraulics of upper technology levels use load-sensing systems like those of Fig. 7.40.

Their first application in the tractor sector was presented by Allis-Chalmers in 1973 in the models 7039 and 7050. The publication [7.35] contains a circuit diagram and detailed data on this development; however, had no followers for a certain time.

In 1987, CASE IH presented its Magnum 7100 series, first in the US [7.42] with 97/112/125/145 kW PTO power and 1989 in Europe. The hydraulic circuit is still typical today in several aspects, **Fig. 7.41**. The variable displacement pump (1) delivers up to 110 l/min. The circuit is described in [7.38, 7.43,] based on [7.42].

A tandem gear pump (2) charges the main pump (1) for smooth filling at high rpms, and for cooling and filtering. The working circuit (max. 180 bar) also serves the power steering (8) via the priority valve (3). Valve (4) keeps the difference between load pressure and pump pressure constant at 25 bar. Hence the flow rates through the directional valves 5 to 8 are independent of both the load pressures of the consumers and the pump speed and are at the same time proportional to the open area of the valves (equation 7.7). Hence, the flow is always largely proportional to the spool displacement, ideal for many practical operations. The valves (5) and (6) serve remote control functions on implements.

The maximum pressure of 180 bar is limited by both, the valve (10) and the pump displacement.

Limiting pressure by the pump displacement prevents losses and heat generation. If more than one consumer is supplied via the valves (5) to (8), only the highest load pressure is directed to the pump valve (4) due to the function of the check valves (9), see also Fig. 7.40. Pressure difference at their directional valves is now not controlled by the pump but by compensator valves (11) and/or (12).

A continuous flow can be adjusted by the valves (13). The relief valve (14) safeguards the maximum LS pressure; the valves (15) serve for automatic unblocking of valve (5) or (6).

The German Agricultural Yearbook Vol. 5/6/7/8 (1992/93/95/96), edited by *Matthies* and *Meier* (bilingual German-English), contains further LS circuits of large tractors of Ford, Fendt, John Deere, and Massey Ferguson. John Deere replaced its constant pressure system in 1992. (The numbering of the yearbook was modified in 1995, therefore no issue is dated “1994”).

Research results on tractor hydraulics were presented in Germany under *Matthies, Harms, Frerichs* [7.36, 7.40, 7.41, 7.43–7.48], *Murrenhof* [7.49, 7.50], *Geimer*, and others.

Research supported developments of piloted valves with digital electro-hydraulic actuation, ready to be connected with ISOBUS systems. **Further innovation potential** is seen in:

- Reduced energy losses by reduced or eliminated control pressure Δp
- Reduced energy losses by circuits with more than one working pressure level.
- Improved control dynamics
- Improved digital control strategies
- Digital hydraulics
- Independent metering valve groups
- Energy regeneration
- Hydrostatic power transformer

Reliable economic electronics are needed to realize innovations in these directions.

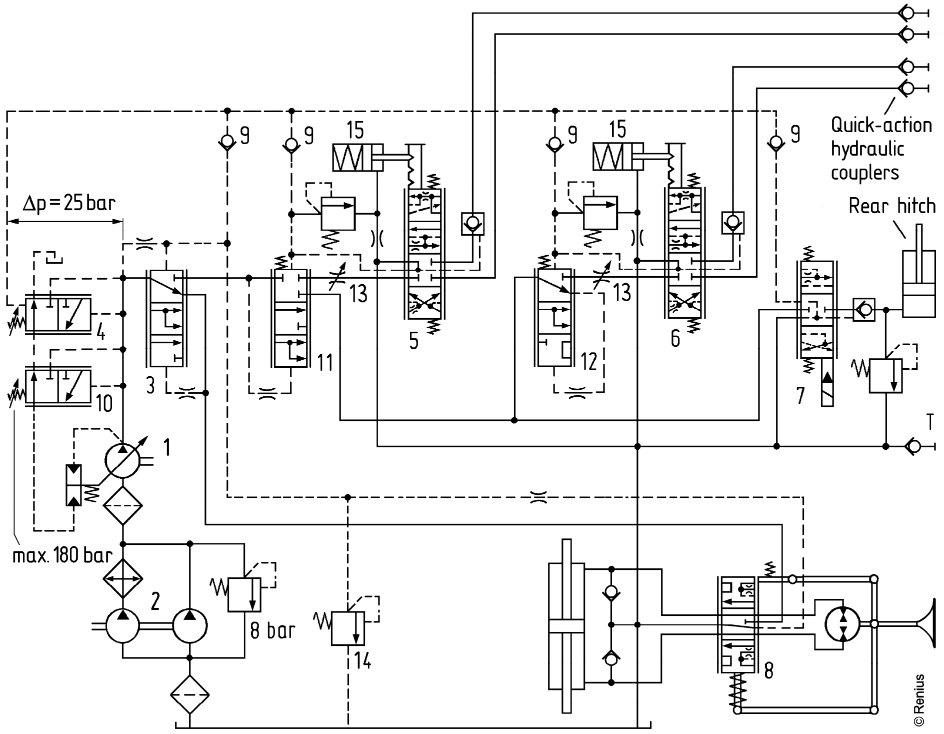


Fig. 7.41: Load-sensing tractor hydraulics of the CASE IH “Magnum” 7100 series, introduced in USA in 1987 and in Western Europe in 1989 (Production from 1988). This circuit initiated a general worldwide breakthrough of load-sensing systems with variable displacement pumps, usually swash plate design (principle see Fig. 7.40). This diagram is based on [7.42] and [7.43]

Energy losses of remote circuits can be reduced by “power beyond”, **Fig. 7.42**, standardized in ISO 17567 [7.38, 7.51].

Implements are usually supplied with hydraulic tractor power via quick-action couplers and controlled by the tractor directional valves. It can make sense to have the valve on the implement.

In such a case, oil flow from the pump should not pass twice through a directional valve to prevent unessential losses, leading to the following principle:

A direct line to the implement bypasses the tractor directional valve, which is now placed on the implement together with the pressure compensator.

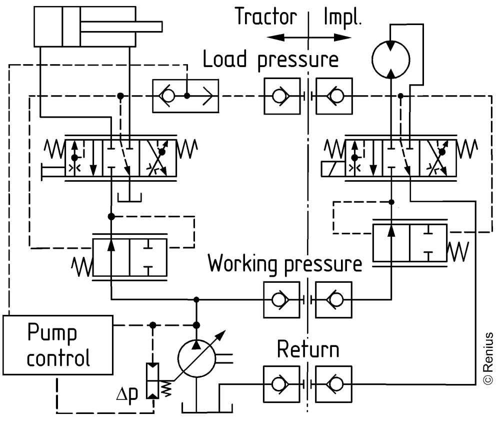


Fig. 7.42: “Power beyond” principle: a direct line to the implement bypasses the tractor directional valve, now located on the implement [7.38, 7.51]

7.7.4 Pumps, remote control valves, and circuit efficiencies

There are four **basic types of pumps** used for tractor hydraulics, **Table 7.20**. Single pumps (low displacement) are adequate for small tractors of technology level I without power steering. Tractors of technology level II can have power steering, using one or two gear pumps (Fig. 3.47 and 3.48), also in a tandem configuration (Table 7.20), which reduces initial costs.

Technology level III is typical for both, low-cost tractors with gear pumps and powerful tractors with variable displacement pumps and LS circuits.

LS systems are standard in levels IV and V.

Table 7.20: Typical pump types for tractor hydraulics and their data as offered by Bosch Rexroth

Pump type	Maximum pressure (tractors)	Typical displacements for tractor applications, cm³	Maximum speed, rpm *)	Data source
Single external gear pump	At least max. 250 bar, but limited for implements to 205 bar (20.5 Mpa, ISO 10448E:2018)	8 11 14 16 19 22	Size 8: 4000 Size 11: 3500 Others: 3000	Bosch-Rexroth Series F
As above, but in "tandem" configurations		Typical examples for tractors: 11-8; 14-11; 16-14; 19-14; 22-14		
Low-noise external gear pump		12 14 16 19 22	Size 12: 3550 Others: 3000	Bosch-Rexroth F "Silence Plus"
Variable displacement swash plate pump		28 45 63 85	Size 28: 3200 Size 45: 2900 63 & 85: 2700	Bosch-Rexroth A10VNO 5x

*) The minimum pressure at input, as defined by the manufacturer, has to be observed ©Renius

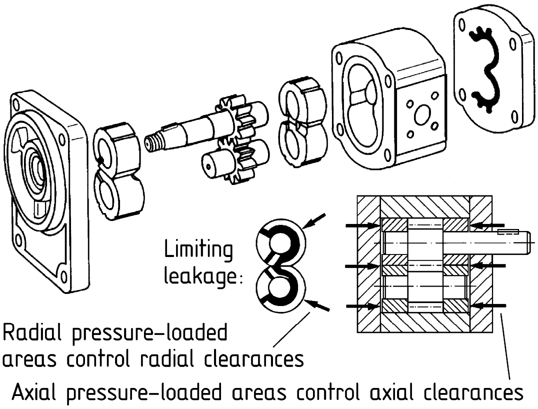

Limiting leakage:
Radial pressure-loaded areas control radial clearances
Axial pressure-loaded areas control axial clearances

Fig. 7.43: Gear pump with clearance controls through pressure loaded seal faces. Courtesy Bosch

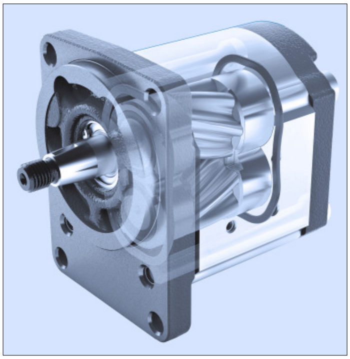
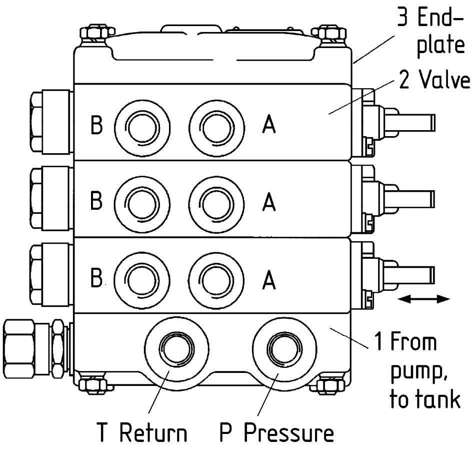


Fig. 7.44: Gear pump "Silence Plus". Courtesy Bosch Rexroth

Modern gear pumps can develop high efficiencies only if their leakage is kept low by axial and radial pressure loaded seal faces, as shown by arrows in the design of **Fig. 7.43**.

The pump of **Fig. 7.44** has a very low noise level, achieved by helical gears with special tooth profiles and low tolerances.



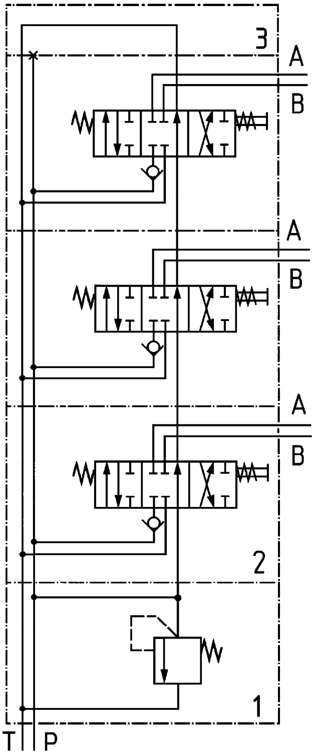


Fig. 7.45: Typical block of manually operated remote control valves for hydraulic open-center circuits. The number of valves is flexible due to uniform interfaces. Only one consumer can always be supplied by A and B. Courtesy Bosch

Open-center circuits for lower technology levels with constant displacement pumps (Fig. 7.33–7.35) normally use simple manual remote control valves, **Fig. 7.45**.

Floating positions are possible by a u-turn shortcut of output A-B (mid position) or by a valve with four positions, one with A-B output shortcut.

Variable swash plate axial piston pumps are popular for load-sensing systems, **Fig. 7.46**. This type from Bosch Rexroth is applied in very high production volumes. The moderate nominal working pressure enables large piston diameters.

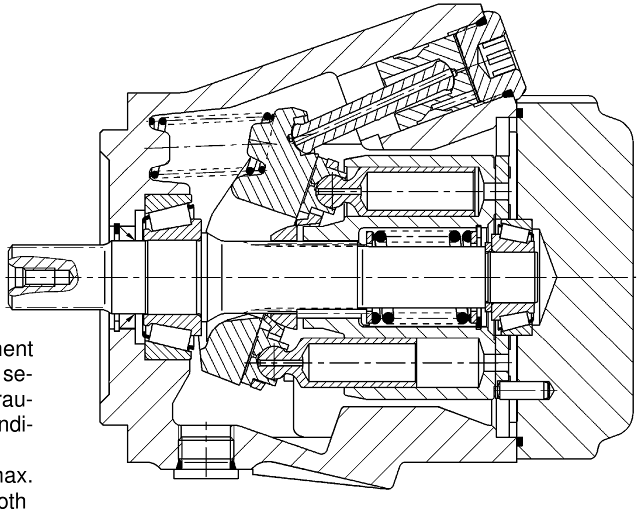


Fig. 7.46: Variable displacement axial piston swash plate pump, series A10 VNO for tractor hydraulics. Sizes 28/45/63/85 (2019) indicating displacement in ccm/rev. Drawing: Size 45, 7 pistons, max. 18.4°. Courtesy Bosch Rexroth

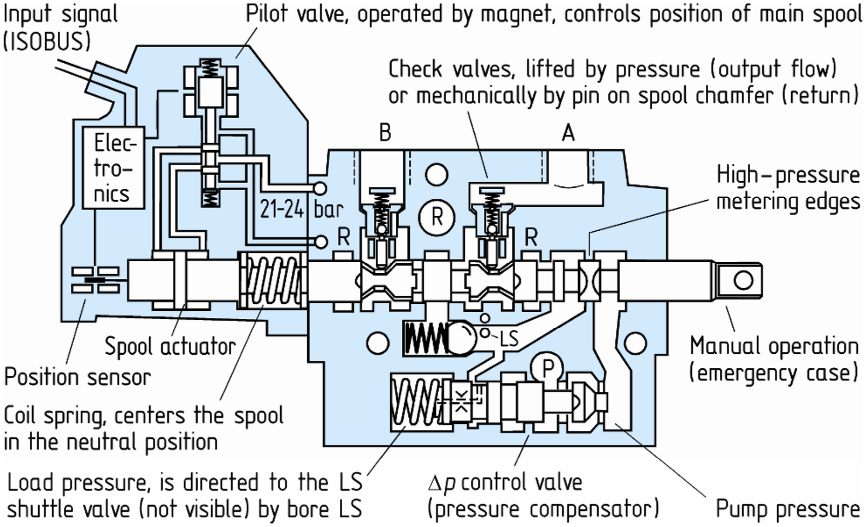

Input signal (ISOBUS)
Pilot valve, operated by magnet, controls position of main spool
Check valves, lifted by pressure (output flow) or mechanically by pin on spool chamfer (return)
B
A
Elec-tro-nics
21-24 bar
R
R
R
High-pressure metering edges
LS
P
Spool actuator
Position sensor
Manual operation (emergency case)
Coil spring, centers the spool in the neutral position
Load pressure, is directed to the LS shuttle valve (not visible) by bore LS
Δp control valve (pressure compensator)
Pump pressure

Fig. 7.47: Electro-hydraulic proportional valve Bosch Rexroth SB23LS-EHS for load-sensing circuits. Nominal flow 80 l/min, max. 140 l/min. Two poppet check valves with almost zero leakage (for mounted implements including front end loader), more in detail in Fig. 7.49.
Drawing: author, based on [7.46] and material from Bosch Rexroth

Typical remote control valves for premium tractors with load-sensing and variable displacement pumps (Fig. 7.46) work with three inputs P, R, and LS. The valve of **Fig. 7.47**, plotted in a simplified cross section, was the first generation of electro-hydraulic remote control valves from Bosch Rexroth; however, produced in large volumes. Its functions are represented by ISO 1219 symbols in **Fig. 7.48** and its technical specifications listed in **Table 7.21**. This size is typical for the lower end of nominal variable pump flows of about 100 l/min.

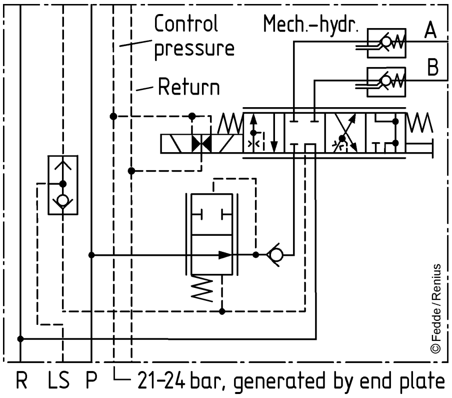


Fig. 7.48: Circuit diagram of the inner elements of the Bosch Rexroth valve of Fig. 7.47 [7.46]

The simplified cross section of Fig. 7.47 shows an actuation of the main spool by a pilot pressure system, which offers better dynamics than a large magnet along with smaller dimensions. The load pressure signal is picked up as indicated by “LS”.

Table 7.21: Specifications of the Bosch Rexroth valve of Fig. 7.47 [7.46]

Valve type:	SB23-LS EHS
Nominal flow	80 l/min
Maximum flow	140 l/min
Nominal pressure	250 bar
Main spool diameter	18 mm
Main spool hub	+/- 7 mm
Axial spool force	max. +/- 200 N
Pilot pressure	21–24 bar
Maximum pilot flow	about 0.7 l/min
Spool dia. Δp-valve	18 mm

In Fig. 7.47 and 7.48, the hydraulic pilot valve typically gets its input signal from a digital ISOBUS line via adequate electronics being able to "read" the digital bus signal and to convert it to an electric flux activating the magnet. Output of this pilot system is a pressure difference directed to the piston actuator of the main spool. The position sensor gives its feedback to the electronics within a *closed loop control*.

A full stroke of the main spool needs only 70 ms, which is sufficient as the analog pump control needs more time.

When the pilot valve pressure moves the main spool to the left, a flow from high pressure input P passes the Δp control valve arriving at the main metering edge.

As this is opened, the flow proceeds to the ball-type check valve and further to the check valve of output A, which is automatically opened by the working pressure.

At the same time, the left check valve at port B is lifted hydro-mechanically, which is easier to understand from **Fig. 7.49**. The lower pin end is actuated by the spool ramp and thus opens in the first phase the small ball-type check valve. This reduces the pressure difference and hence the necessary lifting force to allow the second phase of a complete opening by the pin shoulder.

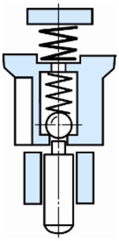


Fig. 7.49: Check valve of the design in Fig. 7.47

Return flow is opened in two phases:

I Pushing the ball for lowering the pressure difference

II Pushing the poppet for complete opening

Valve variants and improvements [7.29]:

- Versions with only one check valve.
- Replacement of the mechanical actuation of the check valve pins by small hydraulic actuating cylinders.
- Integration of a Δp return flow control in the check valve to make the flow independent of the return flow pressure.
- Removing the right end of the spool, renouncing the possibility of manual operation (trusting in system reliability), saving space and initial costs.

Spool actuation by electric stepper motor is an alternative principle, **Fig. 7.50**, which has some advantages such as

- No need for a pilot pressure
- No need for a position sensor as position is derived from electric motor control
- Low energy consumption
- Less sensitive to oil contamination

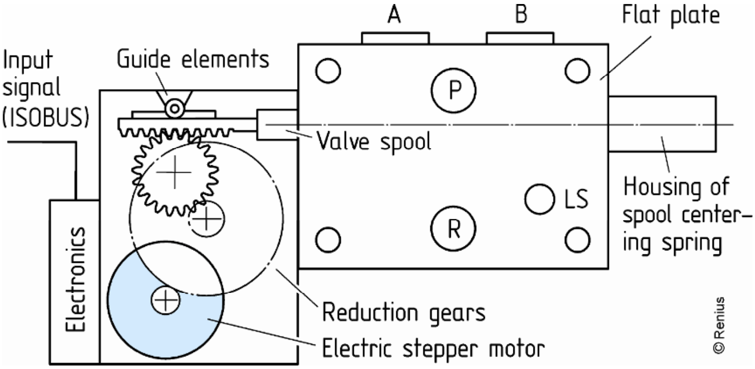


Fig. 7.50: Basic structure of a proportional spool valve actuator consisting of an electric stepper motor and high-ratio speed reduction gears. Similar design is applied, for example, by John Deere

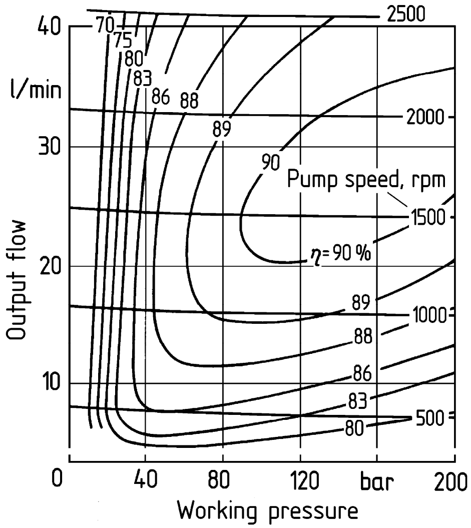


Fig. 7.51: Efficiencies η of a gear pump [7.52]

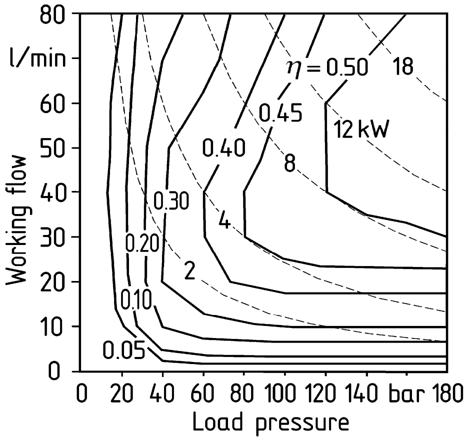
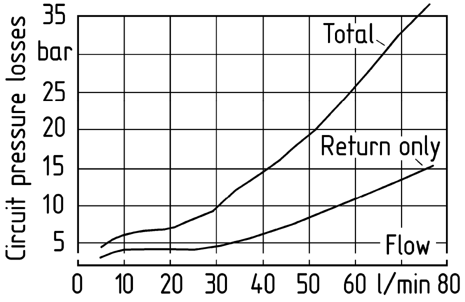


Fig. 7.52: Total efficiency of a complete hydraulic tractor - implement power transfer [7.46]



Efficiencies of driving implements by hydraulics or electrics. Hydrostatic power transfer is very popular, in particular for premium tractors. Typical nominal output flows are 90–120 l/min for LS systems with variable pumps, but can even be much higher for large tractors. The popularity is probably a result of the very high flexibility along with moderate initial costs.

Modern gear pumps (as shown in Fig. 7.43 and 7.44) can offer high efficiencies in spite of their simple and cheap design, **Fig. 7.51** [7.52]. Modern electric generators have even better values but their costs are higher and their outside attachment to diesel engines or PTO powertrains rather more complex; however, not impossible.

Swash plate pumps as shown in Fig. 7.46 have similar efficiencies; the same for most other types of hydraulic motors. Low-cost/high-torque motors such as the “Orbitrol” from Danfoss have lower efficiencies but save space and additional gear reductions.

Practical losses of a typical hydraulic circuit when driving a hydraulic motor on an implement (for example a blower on a pneumatic seeding machine) by load-sensing tractor hydraulics were published by *Fedde* [7.46], **Fig. 7.52**. The power transfer is done by a variable axial piston tractor pump, which is fed by a charge pump and controlled by an LS valve. The pump flow is directed through pipes, valves, hoses, and quick couplers to a hydraulic motor on an implement, and the return flow is directed back to the tractor oil reservoir.

The *best total efficiency is only 50%*, explaining the need for intensive cooling. Pressure measurements indicated high losses only for fluid circulation, **Fig. 7.53**. Large return pipe diameters and circuits *without charge pumps* are recommended.

An electric power transfer might be able to cut such total losses by about half.

Fig. 7.53 (left)**:** Measured circuit pressure losses for the results of Fig. 7.52 [7.49]

7.7.5 Connections, filters, and oils

Precision steel tubes serve to transfer hydraulic power and control signals.

Maximum nominal pressure for tractor-implement power transfer is standardized in ISO 10448 [7.16] at 205 bar. This value dominates all hydraulic tractor circuits and hence also the maximum pump pressure, typically 210 bar for the total on-board net.

However, peak pressures above 210 bar are possible in pipes between a closed valve and a cylinder, for example, by mass forces of mounted implements such as heavy plows or front end loaders. Special "shock" pressure relief valves are common near the cylinder (opening, for example, at 220–240 bar) to safeguard the structures and the hydraulic system.

Steel tube dimensions should be based on two criteria:
- Average fluid velocity (pressure losses)
- Wall thickness (stability)

Calculation of inner diameter d of a pipe with an average section velocity v and flow Q yields

$$d\,[\text{mm}] = 4.607\cdot\sqrt{\frac{Q\,[\text{l/min}]}{v\,[\text{m/s}]}} \tag{7.6}$$

Example: The inner diameter d of a high-pressure tractor working pipe shall be calculated for a nominal flow of 100 l/min at 210 bar nominal pressure. An average fluid velocity of max. 5.5 m/s is recommended in [7.38] for continuous operation.

Equation 7.6 yields d = 19.64 mm leading to an inner pipe diameter d = 20 mm.

Addressing the necessary wall thickness of the pipe, ISO 10763:1994 [7.53] states the following formula for the static burst pressure p_b of a pipe:

$$p_\text{b}\,[\text{bar}] = 10\cdot R_\text{m,\,min}\,[\text{N/mm}^2]\cdot\left(\ln\frac{D}{d}\right) \tag{7.7}$$

where

$R_\text{m,\,min}$ = minimum tensile material strength, e. g. 360 N/mm^2 (ISO 10763)
D, d = tube outside and inside diameters

Example: Diameter D = 25 mm, d = 20 mm, steel 1.0255 (St 37.4), $R_\text{m,\,min}$ = 360 N/mm^2 (as recommended in ISO 10763):

$$p_\text{b} = 803\,\text{bar}.$$

ISO 10763 recommends a nominal pressure of ***1/4 of burst pressure*** *to cover pressure peaks and safety,* ***Table 7.22****.*

But at the same time ISO 10763 allows modifying the factor ¼, defining it as subject to adjustment between the manufacturer and user according to actual dynamics.

In the example, a maximum pressure of the relief valve of 240 bar would be 30 % of the calculated burst pressure. This is slightly more than 1/4, but can be tolerated due to the pressure peak limitations by the mentioned shock valves.

A standardized German calculation method for *dynamic pressures* is offered by DIN 2413 using S-N curves.

An endurance limit of 225 N/mm^2 is cited for steel 1.0255/St 37.4 for pulsating pressure (from lower value near zero to upper value, etc.). The related S-N curve allows cumulative damage fatigue estimations.

Table 7.22: Burst pressures and nominal pressures for tubes of steel 1.0255 (St 37.4) based on equ. (7.7), ISO 10763. Required minimum tensile strength: 360 N/mm^2 (ISO 10763) [7.54]

		Tube wall thickness, mm					
		1	1.5	2	2.5	3	4
Tube outside diameter, mm	10	803 201	1284 321	1831 460	— Burst pressure — Nominal pressure		
	12	656 164	1036 259	1460 365	1940 485		
	15	515 129	803 201	1117 271	1460 365	1839 460	
	18	424 106	656 164	905 226	1172 293	1460 365	
	20		585 146	803 201	1036 259	1284 321	
	25	300 75	460 115	628 157	803 201	988 247	1388 347
	30			515 146	656 201	803 259	1116 321
	35		323 81	437 109	555 139	677 169	934 234

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Flexible high-pressure hoses enable remote control of and power transfer to implements. The pressure forces are mainly carried by a steel matrix. A burst can cause severe injuries. A general design rule is:

Prevent any additional stress, which can, for example, be generated by low curvature radii (bending) or additional outside forces.

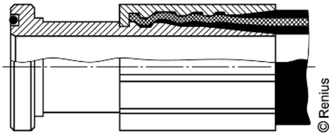

Fig. 7.54: Hose fitting by pressing with radial deformation. Flange ISO 6162 (SAE J518)

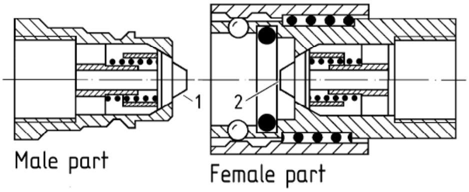

Fig. 7.55: Design of a coupler unit [7.38]

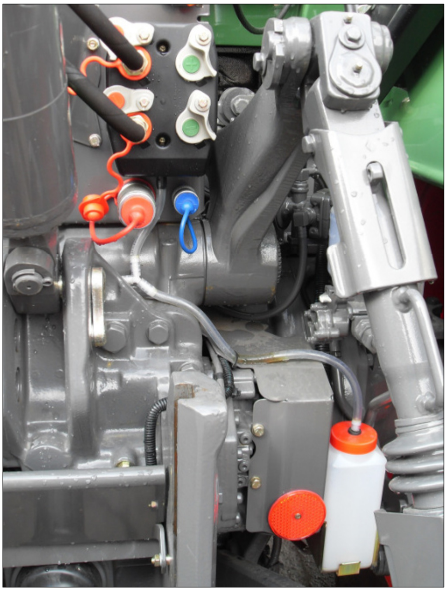

Fig. 7.56: Spill container Photo: Author

The inside diameter is the nominal value. Durability should be carefully tested under cyclic pressure loads and additional flexing, based on ISO 6802 [7.55].

The connection between tube and fitting can be a typical weak point; hence, modern mounting with a special pressing tool is recommended, **Fig. 7.54**.

Hoses increase the hydrostatic capacity of hydraulic circuits reducing its stiffness; however, improve damping of vibrations [7.38, 7.50], which is usually welcomed.

Quick couplings connect the tractor hydraulics with implement circuits [7.56].

ISO 5675 [7.57] states:

The coupler shall comply with the operating requirements of ISO 7241-1 series A and the requirements of ISO 5675.

Fig. 7.55 shows both parts in the disconnected mode. If they are connected, poppets 1 and 2 open one another.

ISO 5675 recommends **nominal diameters of 12.5 and 20 mm,** corresponding to the recommended nominal inside diameters of the hoses used. The maximum pressure is limited to 250 bar; the pressure drop should not exceed 3.5 bar at 45 l/min (for size 12.5) and at 70 l/min (for size 20).

Further recommendations define the location at the tractor, the forces for connecting and disconnecting, and the spill; for example, 12.5 ml when disconnecting under a pressure of 175 bar. This volume is not high, but better to collect it in a small spill container, **Fig. 7.56**, keeping the soil and the tractor free of oil.

A second standard ISO 16028:2015 [7.58] defines flush type coupler versions with rated flows up to 189 l/min and very low pressure losses of only 1 bar. Nominal diameters are: **6.3/10/12.5/16/19/25 mm**.

These dimensions of ISO 16028 are not compatible with those of ISO 5675. However, these couplers are useful for large tractors with remote flows ≥ 100 l/min.

Metallic tube fittings enable the economical mounting and dismounting of tubes. One basic principle is shown in **Fig. 7.57**.

This initial design uses a hardened ferrule (black) with one cutting edge which is pressed into the 24° cone by tightening the cap nut with a defined torsion moment, typically corresponding with about one "fixing turn" of the nut. As the tube is usually made of mild steel, the ring can cut a small groove into the tube surface, at the same time creating a small wall supporting the sealing (right part of Fig. 7.57).

Such a design has the disadvantage that the functions "sealing" and "carrying external forces or moments" are not separated.

This can create leakage, mainly under vibrations with dynamic bending moments. A simple test setup, **Fig. 7.58**, demonstrates the problem and can be used for a first evaluation of design improvements. However, detailed test methods are standardized in ISO 19879 [7.59].

Modern versions as standardized in ISO 8434-1 [7.60] keep the uniform cone angle of 24°; however, now recommend ferrules with two cutting edges, **Fig. 7.59**. The broader base offers improved stability against vibrations; see also the commercial example in **Fig. 7.60**. In addition, vibrations can also be reduced by elastic tube clamps.

If all this is not sufficient, the two functions of "sealing" and "carrying loads" should be separated as shown by example of **Fig. 7.61.** This design has outstanding reliability and a low risks of incorrect mounting but needs a special machine for preparing the tube and is slightly more expensive than a standard fitting.

Another also very reliable design uses fittings for flared tubes (angle 37°, ISO 8432-2:2007). Welded fittings are also reliable but even more expensive.

All fittings should be able to handle a nominal pressure of 25% burst pressure.

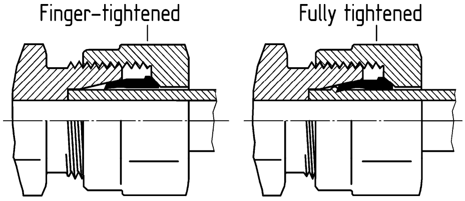

Fig. 7.57: Simple initial design of a tube fitting (DIN 2353) with "only one edge" ferrule

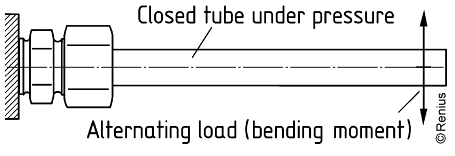

Fig. 7.58: Testing resistance against leakage under inside pressure and external vibration

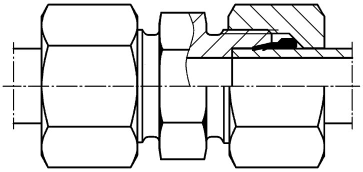

Fig. 7.59: Metallic tube fitting of ISO 8434-1: Ferrule with two cutting edges

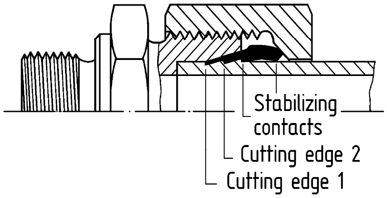

Fig. 7.60: Commercial example: tube fitting "EO-Plus" from Parker Hannifin

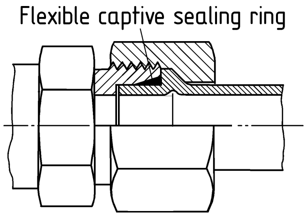

Fig. 7.61: Robust fitting WALFORM with captive sealing ring. Courtesy Eaton's Walterscheid

Hydraulic filters reduce contamination [7.61], which is “responsible for the majority of system breakdowns” (Bosch).

Oil contamination can create

- Higher *wear* of components under load
- *Abrasion* at points of high fluid speed
- Earlier *fatigue* of rolling bearings
- *Dysfunctions* of servo valves
- *Clogging* of small orifices

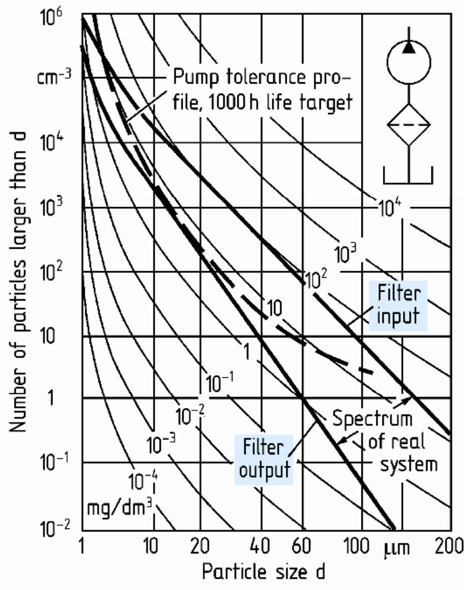

Fig. 7.62: Cumulative distribution of particles by numbers and concentrations in hydraulic fluids. Inserted: Pump tolerance and filter effect [7.63]

Table 7.23: Particle codes of ISO 4406 (extract)

ISO code	Particles in 100 ml from	to
20	500,000	1,000,000
19	250,000	500,000
18	130,000	250,000
17	64,000	130,000
16	32,000	64,000
15	16,000	32,000
14	8,000	16,000
13	4,000	8,000
12	2,000	4,000
11	1,000	2,000
10	500	1,000

Contamination research results of the Oklahoma State University by *Fitch et al.* [7.62] were extracted for agricultural machinery hydraulics by *Böinghoff* [7.63].

Developed contamination profiles can represent actual contamination and given limits of hydraulic components, **Fig. 7.62**.

The special log-based scales of the diagram enable almost linear distributions of real system profiles in a cumulative manner. The smaller the particles, the larger their number, with a progressive increase.

Example to read the diagram:

There are 3500 particles of 20 µm *or larger* in one cm^3 fluid at the filter input corresponding with 200 particles at its output. In total, the filter is able to reduce the particles for meeting the tolerance profile of the pump (example for 1000 h life).

Several international standards have been developed for evaluating system contamination and filter performance:

- *Counting particles* of a given fluid automatically and processing the data by statistical methods, ISO 11943 [7.64].
- *Evaluating filter performance* for given particle sizes by comparing upstream and downstream contaminations: multi-pass or beta test, ISO 16889 [7.65].
- *Coding particles*, ISO 4406 [7.66].

Codes of counting results (ISO 4406) are extracted in **Table 7.23**. A typical requirement for mobile hydraulics is **19/17/13**, which means, that 100 ml contain

- max. 500,000 particles **≥ 4 µm**
- max. 130,000 particles **≥ 6 µm**
- max. 8,000 particles **≥ 14 µm.**

The filter test of ISO 16889 results in a *beta value* based on the following relation:

$$\beta_x = \frac{\text{No. of particles} > x\,(\mu\text{m})\ \text{upstream}}{\text{No. of particles} > x\,(\mu\text{m})\ \text{downstream}} \quad (7.8)$$

Giving an example, we can derive from Fig. 7.62 a value of $\beta_{20} = 3500/200 = 17.5$.

Filter requirements are listed in **Table 7.24**. They are valid not only for hydraulics but also for transmissions (and of course engines). Fineness should be based on the ISO coding as shown above.

Simple transmissions have no oil filter systems but those of upper technology levels have them, often in auxiliary pressure circuits for shift actuators, cooling, and lubrication.

There are two basic filter designs:

1. Mech. filters (“screen, mesh”)
2. Absorbent filters (“capillaries”)

Absorbent filters are predominant because of their low costs and high deposit capacity, **Fig. 7.63** (left) while mechanical filters offer a better filter effect above their mesh size (right). They are, for example, installed as fixed-mesh suction filters or fixed-mesh high-precision filters upstream of pilot-operated directional valves.

Filters for hydraulics, transmissions, and engines are usually of type 2. Efficiency of contamination control also depends on the filter position in the circuit, **Table 7.25**.

Table 7.24: Basic requirements of filter specifications

- Adequate fineness for the problem
- High filtering performance: large β values
- High capacity of contamination storage
- Burst pressure according to the circuit
- Low pressure losses Δp
- Low Δp increase during service
- Bypass valve preventing destruction
- Sensor indicating end of service life
- Easy to replace without tools
- Low fluid loss in case of replacement

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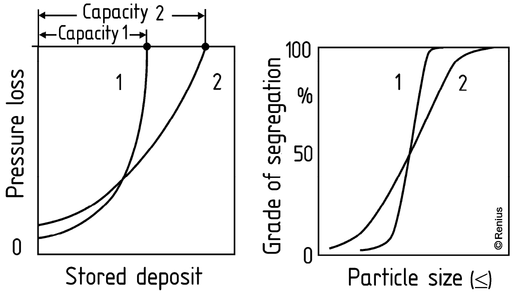

Fig. 7.63: Filter performance: 1 mechanical, 2 absorbent

A typical configuration is the combination of a suction filter and a return flow filter which should also collect leakage flows as they are usually highly contaminated.

High-pressure filters offer the best results but are expensive, heavy, and seldom applied for tractors.

Table 7.25: Oil filter systems. Suction filters (for larger particles) often combined with others [7.38]. Leakage of pumps is usually more contaminated than the main flow, should be filtered directly

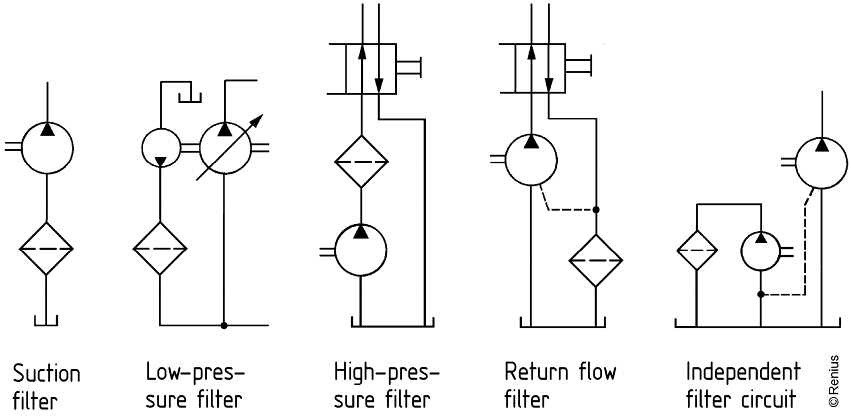

Table 7.26: Comparing transmission and hydraulic oil circuits; advantages indicated by +

Oil circuits:	Combined	Separated
- Logistics	+	
- Common components	+	
- One or two coolers	+	
- Costs of piping	+	
- Total initial cost	+	
- Risk of contamination		+
- Reserve of oil volume		+
- Appropriate specifications		+
- Temperature limits		+
- Transmission performance		+

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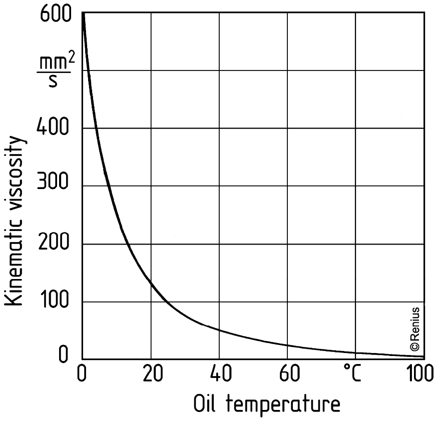

Fig. 7.64: Kinematic viscosity versus temperature for typical hydraulic oil (HLP 46, VI =100)

Oil specifications have an important influence on the wear and losses within a hydraulic system. First of all, the designer must decide whether the circuits of the tractor hydraulics and the transmission shall be combined or not. Both systems are used and have pros and cons, **Table 7.26**.

The most important oil property is its viscosity (see also chapter 5.7.2).

High viscosity supports surface separation by oil films (reducing wear) but increases losses at high sliding speeds. This conflict requires a compromise, which is usually improved by additives reducing the wear under mixed-friction conditions.

The viscosity of hydraulic oils greatly depends on temperature changes, **Fig. 7.64**, which can only slightly be influenced by additives (e. g. by large molecules).

Common hydraulic oils typically have a nominal kinematic viscosity of 46 mm²/s as defined in ISO 3448 for 40°C, **Fig. 7.65**, bold line. Other values may be needed for extremely cold or hot climates.

An upper limit of 80 °C is recommended for tractor hydraulics.

UTTO oil (**U**niversal **T**ractor **T**ransmission **O**il) is used for common oil circuits, **STOU** includes the engine (seldom used).

As an example of UTTO, John Deere offers its "Hy-Gard", which has a nominal kinematic viscosity of 59 mm²/s, hence between typical hydraulic and transmission oils, and is able to handle the specific demands of wet brakes. **Engine oils** can meet hydraulic oil *viscosity* demands, but other properties are not perfect.

Disposable tank reserves of oil volume (e. g. for lift cylinders of large dumper trailers) should be about 30–80 l (50–300 kW) in Europe. This can require extra tanks.

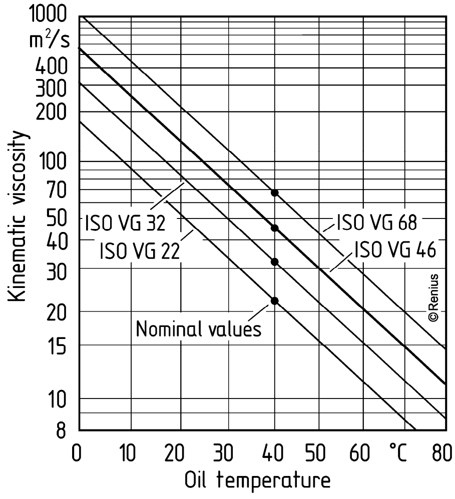

Fig. 7.65 (left)**:** Ubbelohde diagram for hydraulic oils with viscosity index VI = 100 [7.38], which is a typical value for the class HLP (DIN 51524) corresponding with class HM of ISO 6743-4

7.8 Tractor ballasting

Agricultural tractors should be designed to be as light as possible to offer high payloads.

Mass-to-power ratios of tractors in Europe were published in 2003 [7.67], **Fig. 7.66**. Actual values may be slightly lower as now generally related to EG power definitions (fans excluded, power values higher than with fan).

If a tractor shall be able to develop high pull forces and traction power at low speeds, it usually needs ballast.

The question is: **How much ballast?** A guideline can be developed by applying the fundamentals of the previous chapters, first by calculating the available power P_{axle}:

$$P_{\text{axle}} = (P_{\text{nom}} \cdot \lambda - P_{\text{aux}}) \cdot \eta \qquad (7.9)$$

where

P_{nom} = nominal engine power (ISO)
P_{aux} = auxiliary power including fan
λ = utilization factor (power use)
η = transmission efficiency

Auxiliaries may absorb 10% P_{nom}. Powertrain management may enable utilization of 90% nom. power, $\lambda = 0.90$. Transmission efficiency (engine flywheel to axle output) may be 85% for the field working range of 5 –15 km/h and the high utilization factor.

Under these conditions, axle power equals 68% of nomimal ISO power. The tractor is ballasted well if axle power is transferred to pulling power with acceptable efficiency. Equation (2.8) from chapter 2 yields for 4WD:

$$P_{\text{axle}} = (\kappa + \rho) \cdot \Sigma F_{\text{W}} \cdot v_0 \qquad (7.10)$$

where

κ = eff. total coefficient of net traction
ρ = eff. total coeff. of rolling resistance
ΣF_{W} = sum of vertical axle loads
v_0 = nominal tractor speed (zero slip)

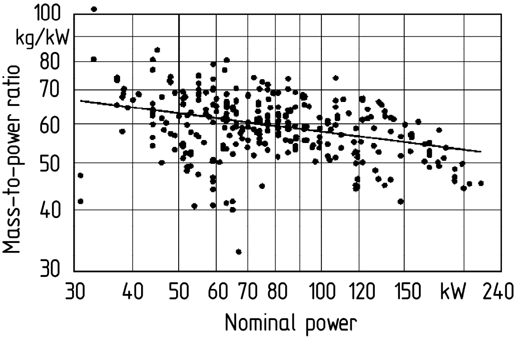

Fig. 7.66: Net-mass-to-power ratios of agricultural tractors with cab and 4WD [7.67], based on dlz data. Practical values often higher due to options

A supposed axle power of 68% of nominal engine power yields the quantity equation:

$$\text{Axle load demand}\,[\Sigma \text{kg}] \approx \frac{250 \cdot P_{\text{nom}}\,[\text{kW}]}{(\kappa+\rho) \cdot v_0\,[\text{km/h}]} \qquad (7.11)$$

Example: P_{nom} = 180 kW, 4WD, dry stubble, given slip 13% → κ = 0.40, ρ = 0.08 (both effective for the tractor), nom. speed v_0 = 7.2 km/h.

→ Equation (7.11) yields **13,021 kg**.

A modern tractor with 180 kW (ISO/EG) may have a net weight of 180 x 55 kg = **9,900 kg**.

Including options, the average practical value is usually higher, for example **10%** *more, which yields* **10,890 kg**. But even this would require additional vertical axle forces of **2,130 kg**. In the case of a rear mounted plow, front ballast of about **1,000 kg** could be right while the rest of **1,130 kg** could be contributed by a proportion of the plow weight and vertical soil forces (see Fig. 7.16 B of this chapter). Tractive efficiency of **72.5 %** (equation 2.9) is near its optimum, see Fig. 2.11 for κ = 0.4, confirmed in [7.68].

This κ–level can be recommended for heavy tillage operations in general as it is within the range of 0.30–0.45 for which the tractive efficiency has always its best values (Fig. 2.11). “Fine tuning” requires real time measuring of actual vertical axle loads to optimize ballast selection and positioning [7.68].

7.9 Front loaders and their consequences for tractor design

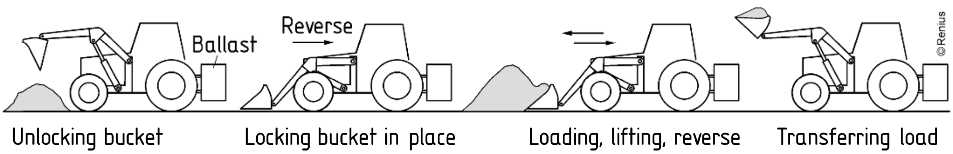

Fig. 7.67: Duty cycle of a low-cost front loader for technology levels I and II, hydraulic cylinders only for lifting. Bucket is locked in place automatically through contacting ground in reverse direction

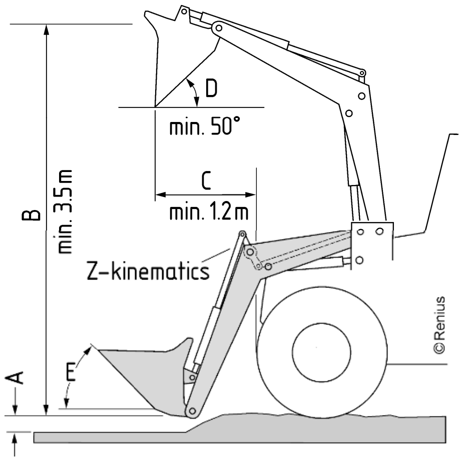

Fig. 7.68: Front loader with active attachment and Z-kinematics for self levelling, typical for premium standard tractors. Symbols: A digging depth, B lifting height, C dumping distance, D angle of emptying, E angle of backwards tilt

Table 7.27: Design preliminaries for a tractor being able to carry a front loader

- Tractor chassis: sturdy frame or additional reinforcement through a frame with a stiff cross beam (Fig. 7.69)
- Tractor front axle and tires must cover much higher loads than for common field work. Special load spectra needed for vertical axle loads and front drive torques. Safety clutch recommended
- Adequate performance of power steering and hydraulic pump flow
- No constraits in serviceability and lights, loader easy to mount and to dismount. Rear ballast available

The **first generation of front loaders** (also called front-end loaders) was simple as the loader attachments (forks and buckets) were only passive: They were locked by lowering the lifting arm to ground contact in reverse mode, **Fig. 7.67**. Such low-cost versions may still be of interest for technology levels I and II for small tractors of nominal 20–50 kW. Recommended payloads are 400–800kg.

Front loaders for tractors of higher tech levels offer more performance and functions (even weighing functions), **Fig. 7.68**, along with higher tractor power, active tool control, driven front axles, LS hydraulics, large tires, and power steering.

A tractor which is able to carry a front loader must have a well prepared chassis with adequate stability, **Table 7.27**.

Preliminaries of front axle dimensioning were already outlined in chapter 3.6.

Adequate **chassis stability** can be realized by the basic structure or through auxiliary frames. Their design should consider recommendations of [7.69], a basic rule for low chassis stresses is shown in **Fig. 7.69**.

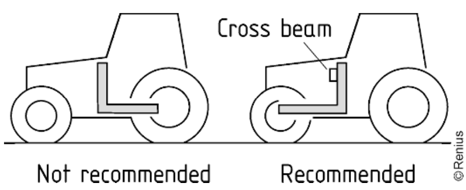

Fig. 7.69: Recommendation of a simple auxiliary frame for front loader applications [7.68]

Premium front loaders offer nearly the performance of wheel loaders, which are also used as mobile farm machines, but are special machines needing special initial investment.

Fig. 7.70 and **7.71** demonstrate the impressive performance of modern tractor front loaders.

Typical **requirements** include a high comfort level with several electronic aids, **Table 7.28**.

Lifting forces (continuously) are about 14 to 30 kN for standard tractors with nominal power of about 50 to 150 kW.

Earthmoving (Fig. 7.71) is the strongest employment and often used for final testing of a new design.

Pressure peaks in the line between the directional valve and the loader cylinder are limited by relief valves; for example, with a nominal setting of 220–240 bar.

The **connection** of loader attachments and lifting arm is standardized in ISO 23206:2005 by defining four connecting points. The standard recommends a maximum tractor net weight of 7.5 t (often exceeded by large tractors) and a maximum lift capacity of 30 kN (common).

The **stability** of a tractor against tipping is reduced considerably when the filled loader lifting arm is in its upper position. Relevant safety aspects are addressed in ISO 16231-1:2013 and DIN EN 12525:2011.

Measurement of a tractor's **center of gravity** (without loader) can be carried out as described in standard ISO 789-6:2019.

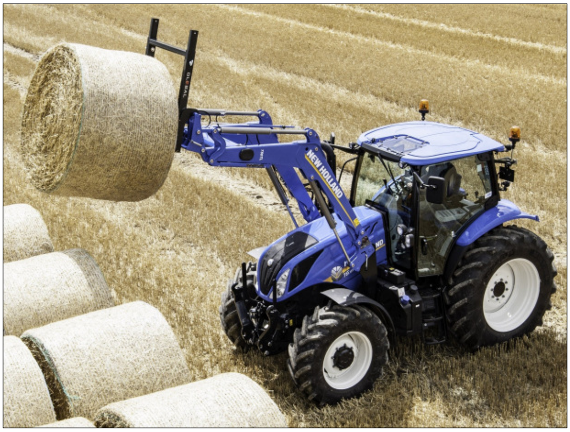

Fig. 7.70: Round baler collecting and loading in the field. Courtesy New Holland (CNH Industrial)

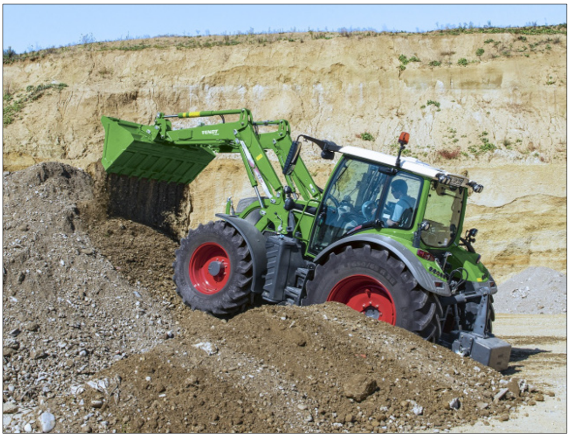

Fig. 7.71: Earthmoving: very heavy work for a front loader, hence, also used for testing durability, handling, and safety. Courtesy AGCO-Fendt

Table 7.28: Requirements of premium tractor front loaders

- Horizontal self-leveling
- Adequate lift capacity
- Convenient steering
- Convenient reversing
- Small turning circle
- Adequate lift speed
- Controlled lowering speed
- Crossgate lever control
- Optimized visibility
- Windshield can be opened
- Integrated damping system
- Weighing and memo functions
- Easy to mount and dismount
- Integrated parking support
- Low weight of the lifting arm
- Warning against tipping

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8 Tractor tests

8.1 Structure of internal tests

Internal tractor test activities can be structured into A, B, C, and D, **Table 8.1**.

They can be organized by teams receiving test orders from the design department and reporting to it or directly within common project management teams.

The experience of the author is that a separation of design and test activities is possible for sections B and D at an early project stage. However, joint teams are recommended at an advanced stage of a project addressing sections A, B, and C, during which related design and test teams should systematically go through the release status of all relevant tractor components at short intervals.

Tests of category B take much time, which can be reduced by lab tests, **Table 8.2**.

Table 8.2: Benefits of lab versus field tests

- Dramatically reduced test times owing to accelerated 24h runs
- High potential of automation
- Reduced costs of tests
- Much better repeatability
- Easier, safer documentation
- Lower noise level
- Higher safety level

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These reasons are very important; however, lab tests require special skills and an understanding of fatigue fundamentals. The alternative of testing tractor components *only* through field operations with complete (usually very expensive) tractor prototypes is today far away from the state of the art and thus not recommended – not even for countries with lower technology levels.

Table 8.1: Possible organization of tests within a modern tractor development process

	Subject	Examples	Type of test
A	Functions, handling	Shiftability of transmission, nominal speeds, tractor handling, ergonomics, electronics, mounting of implements, remote control, hitch forces, fuel consumption, heat balances of all components, . . .	Lab and field tests
B	Endurance of components	Torque loads of transaxle and front axle. Chassis bending and torque loads, loads on axle housings, hitches, suspensions, pumps, . . .	Mainly lab tests
C	Preliminary tests addressing homologation and consumer tests	Cab safety structure, noise levels, engine emissions, braking performance, hand forces (with emergency steering), pedal forces, visibility, cab climate, seat damping performance, safety, consumer tests . . .	Lab and field tests
D	Research	Load and speed spectra, predictive maintenance, electrification . . .	Field measurements

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8.2 Fundamentals of cumulative damage analysis

8.2.1 Introduction and history

Table 8.3: Selected historic milestones

1924 *Palmgren:* First algorithm on cumulative damage, defined for ball bearings [8.1]
1932 *Kloth* and *Stroppel:* First load spectra [8.2, 8.3]
1939 *Gaßner:* First repeated load sequence for lab tests (8 steps), typically for airplanes [8.4]
1945 *Miner:* "Miner rule", applying *Palmgren's* algorithm to other machine elements [8.5]
1970 *Haibach:* Modified *Miner* rule [8.6]

Relevant for *powertrains*:

1940 (about) Torque measurements at drive axles of trucks and passenger cars [8.7]
1950 and later: Publications on measured transmission loads, mainly for tractors [8.8–8.18]. Most fatigue calculations use *"Miner"*
1976 *Renius:* Booklet on fundamentals of load and speed spectra for tractor transmissions [8.19]
1976 *Renius:* Strategy of minimizing dimensions of gear wheels holistically for given load spectra; S-N curves with load scale instead of stress, based on *Miner-Haibach* [8.20]
1977 *Renius:* Sizing transmission machine elements of tractors for given load spectra [8.21]
1990 *Schaller:* Confirmation of *Miner-Haibach* for gear wheel surface contact (pitting) [8.22]
1994 *Suchand:* Confirmation of *Miner-Haibach* for gear wheel tooth bending [8.23]
2002 *Mariutti:* Transmission torque load spectra of a rubber belt tractor [8.24]

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Some fundamentals have already been addressed in previous chapters for rolling bearings, gear wheels, shafts, complete transmissions, and other elements including PTO shafts.

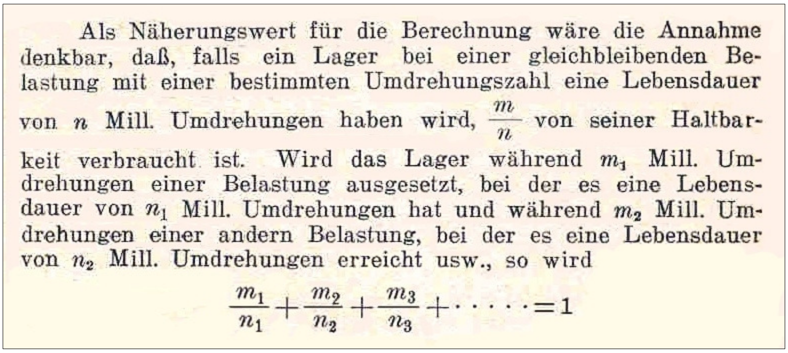
This chapter summarizes the whole process of calculation and testing which requires the following activities:

- Measurement of service loads/speeds
- Statistical treatment of stored data
- S-N curves, scaled in stress and/or load
- Cumulative damage models
- Accelerated tests for verification

Important historic milestones are listed in **Table 8.3** with general publications in the upper part and a focus on powertrains in the lower part.

A. Palmgren (1890-1971) presented the first linear cumulative damage hypothesis in 1924 [8.1]. The extract in **Table 8.4** shows his famous equation (see also equation 5.18 of this book). His approach became the basis for the whole discipline. He initially proposed his algorithm for ball bearings, but it became common for all rolling bearings; with since then some refinements as shown in chapter 5.7 of this book.

Table 8.4: *A. Palmgren's* cumulative damage model, copy from his original publication 1924 [8.1]. m_1 = actual revolutions (of life) for load 1, n_1 = revolutions under load 1 up to bearing failure, etc.



Als Näherungswert für die Berechnung wäre die Annahme denkbar, daß, falls ein Lager bei einer gleichbleibenden Belastung mit einer bestimmten Umdrehungszahl eine Lebensdauer von n Mill. Umdrehungen haben wird, $\frac{m}{n}$ von seiner Haltbarkeit verbraucht ist. Wird das Lager während m_1 Mill. Umdrehungen einer Belastung ausgesetzt, bei der es eine Lebensdauer von n_1 Mill. Umdrehungen hat und während m_2 Mill. Umdrehungen einer andern Belastung, bei der es eine Lebensdauer von n_2 Mill. Umdrehungen erreicht usw., so wird

$$\frac{m_1}{n_1} + \frac{m_2}{n_2} + \frac{m_3}{n_3} + \cdot \cdot \cdot \cdot \cdot = 1$$

In 1932, early statistical fundamentals on service loads were published by agricultural engineers *W. Kloth* and *T. Stroppel* [8.2] who measured torque loads at the input shaft of a binder, driven by the tractor PTO at 540 rpm. The author believes that these authors could be the first ones to apply statistical methods to processing load measurements and to recommend life predictions with S-N curves.

At that time they were mainly interested in peak loads, **Fig. 8.1**, additionally also in time portions. *Kloth* and *Stroppel* first used the term “Kollektiv” in 1936 [8.3].

They stimulated new methods of dimensioning with load spectra not only for agricultural machinery but also for the automotive and aircraft sectors.

Ernst Gaßner (1908-1988) was probably the first to develop repeated load sequences for lab tests, based on load spectra [8.4], in close co-operation with *A. Teichmann.*

In Germany, they both introduced the term “Betriebsfestigkeit” as a special new word which is equivalent to the English term “cumulative damage in fatigue.” Both address calculation and test methods for machine elements or components, in particular for random type service loads.

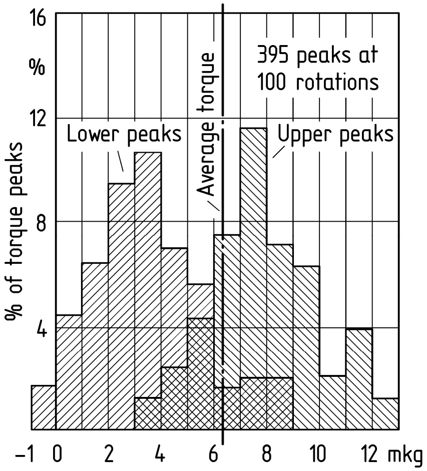


Fig. 8.1: Early statistical analysis of load measurements resulting in a load spectrum (in this form also called “load histogram”), presented in 1932 for a PTO drive to a binder by *Kloth* and *Stroppel* [8.2]. Reconstructed original

The *principle of damage calculation* is shown in **Fig. 8.2** for the classical *Miner* rule [8.5] (left) and *Haibach’s* refined proposal [8.6] (right). Each damage fraction is calculated by dividing the actual cycle by the number of cycles to failure.

Both models require that the sum of all damage fractions must be less than 1.

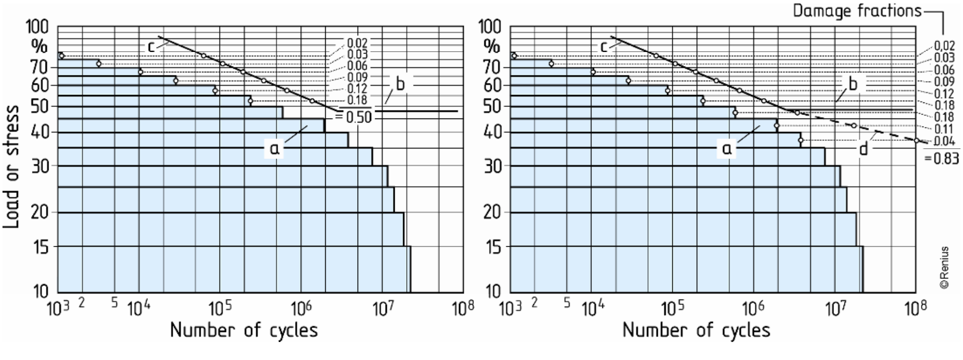


Fig. 8.2: Cumulative damage calculation: Left diagram “Miner original” and right “Miner-Haibach”. **a** load spectrum (histogram = class frequency), **b** fatigue endurance limit, **c** finite life leg, **d** virtual “Haibach” leg with slope -(2k-1) instead of - k (k is the slope of leg c).
Both models allow in principle a *damage sum of* ≤ 1.00. Due to spreading of parameters, several publications recommend damage sums ≤ 0.5 for “Miner-Haibach”. On this basis, the author recommends probabilities for tractor load collectives and load-scaled S-N curves of at least 90%

8.2.2 Load counting methods

Important rule: Statistical counting of measured loads must be adjusted to the type of dimensioning problem, **Table 8.5**.

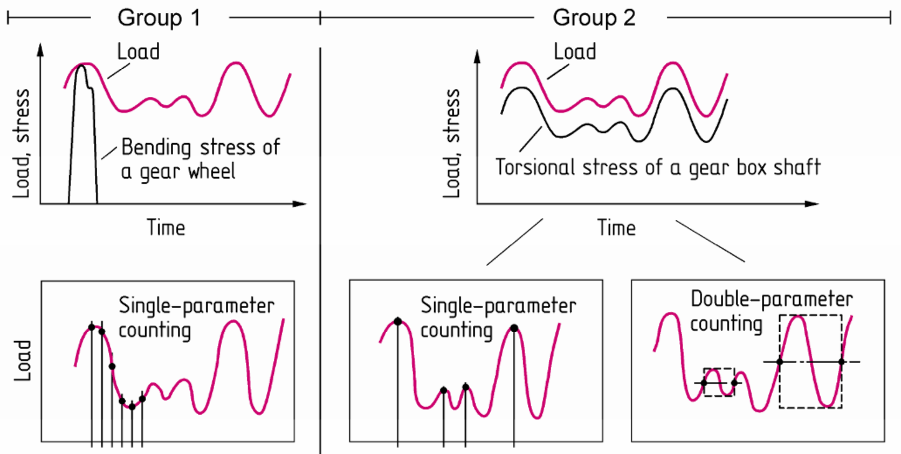
The first group needs triggered sample picking or time portions. This method was, for example, recommended by John Deere already in 1962 [8.12], by ZF in 1973 [8.9], and by the author in 1976 [8.19].

Triggered sample picking is the standard counting method for group 1 load spectra. If the rotational speed is nearly constant, time portions can be used [8.9, 8.19].

Rainflow counting is not appropriate for group 1; however, recommended for group 2.

An explanation of this method and its application to tractors and agricultural machinery was presented in [8.25].

Table 8.5: Adapting the counting method of a load spectrum to the type of dimensioning problem


Group 1
Group 2
Load, stress
Load
Bending stress of a gear wheel
Time
Load, stress
Load
Torsional stress of a gear box shaft
Time
Load
Single-parameter counting
Single-parameter counting
Double-parameter counting

	Group 1	Group 2	
Counting method	**Time portions** Sample picking with constant frequency, results in time portions (very easy with a PC). Triggered sample picking, if rotational speed varies considerably.	**Peak load counting** Best results when the lower load/stress values are near zero. Small amplitudes can be filtered out. The character of the plotted load should be documented. Now often replaced by rainflow counting	**Rainflow counting** Counting of closed cycles. Sorting into classes of mean stress or load. Each class contains an individual spectrum of amplitudes. Special procedure for the “residue”
Load spectrum	2-dim.		...3-dimensional
Examples of application	• Gear wheels • Rolling bearings • Shafts with rotating bending such as axle output shafts	• Shafts (torsion) such as transaxle and PTO shafts • Structures (torsion, bending) such as chassis elements, transmission housings, axle housings, 3-point linkage elements, front loader lifting arms, elements loaded by hydraulic pressure, non-rotating gear wheels (diff.) ...	
Material property	Best basis for “material property” is a load-cycle fatigue curve for a mounted element, calculated from an S-N curve or experienced directly by test runs	S-N curve or load versus cycle fatigue limit curve with $R = \frac{\text{Lower peaks}}{\text{Upper peaks}}$ R derived from load track	S-N curves or load-versus-cycle fatigue limit curves for each class of mean stress or load.

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8.2.3 The practical process

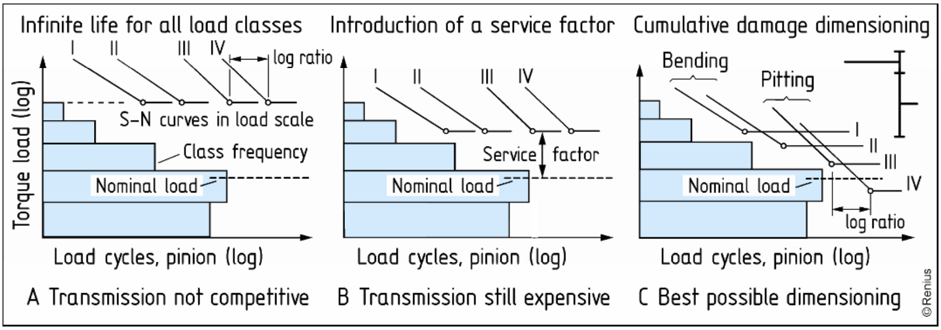
Practical cumulative damage analysis is often carried out on stress levels. This can make sense for problems of group 2 of Table 8.5; however, *not as well for group 1, for which the author clearly recommends load scales* [8.20, 8.21].

Such logarithmically scaled load ordinates simplify the use of fatigue curves of mounted gear wheels, **Table 8.6**. The strategy C on the right of the table is the best at saving costs; hence underlines the benefits of cumulative damage analysis.

Optimized sizing of mounted gear wheels as presented by the author in 1976 [8.20] (Fig. 5.112) was applied to Deutz transmissions from 1975, **Fig. 8.3** [8.26]. The dimensioning diagram of the rear axle bevel gearing of a new Deutz transmission shows three S-N curves in torque load scale in almost ideal positions. Shot peening reduces their statistical spread [8.27].

Forecasting of such torque load-based S-N curves is challenging. Hence, one-level lab tests were run first to verify them. This holistic approach was refined in [8.28].

Table 8.6: Three strategies of dimensioning of a pair of gear wheels, expressed by the position of *load-based* S-N curves for the four main fatigue criteria (Table 5.22). See also Fig. 5.112


Infinite life for all load classes
Introduction of a service factor
Cumulative damage dimensioning
I II III IV
log ratio
Torque load (log)
S-N curves in load scale
Class frequency
Nominal load
Load cycles, pinion (log)
A Transmission not competitive
I II III IV
Service factor
Nominal load
Load cycles, pinion (log)
B Transmission still expensive
Bending
Pitting
I
II
III
IV
Nominal load
log ratio
Load cycles, pinion (log)
C Best possible dimensioning
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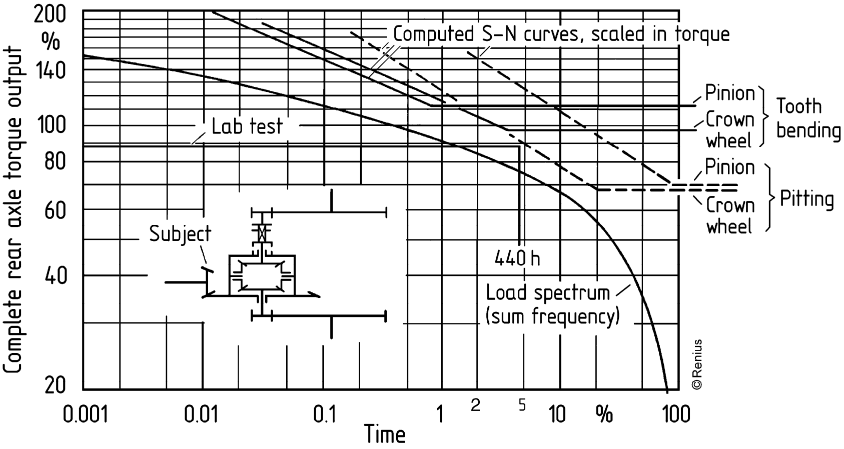

Fig. 8.3: Calculation and test of the bevel gearing of the Deutz transmission TW90 (later TW900) as an example [8.26]. As the ordinate is of torque (instead of stress), test results can be easily inserted and evaluated. A first 440h lab test was undertaken with a constant high load at a typical heavy tillage speed verifying forecasted load-scaled S-N curves of the two *mounted* bevel gears

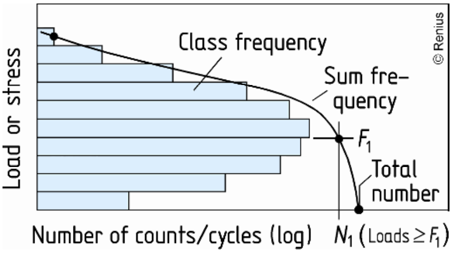

Fig. 8.4: Class frequency and sum frequency

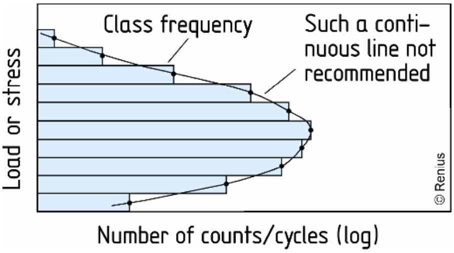

Fig. 8.5: A continuous line through a class frequency can be a source of mistakes as it yields a density distribution only for class width 1 and is not needed for cumulative fatigue analysis

Table 8.7: Acceleration factors for lab tests

A1 Considers removing low loads/stresses
A2 Considers an increased test speed
- in rpm (relevant in group 1, table 8.5)
- in cycle frequency in Hz (relevant in group 2, table 8.5)

A3 Considers disturbing vibrations of the test equipment
A4 Considers high local temperatures

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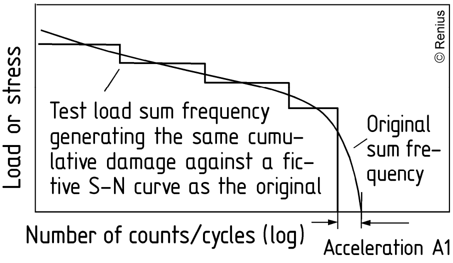


Fig. 8.6: Acceleration factor A1 for transmission lab tests. A1 = real life/test time

Load spectra and lab test strategies

Component testing is usually started before prototypes are available, which needs a fund of representative load spectra.

As shown in chapter 5, dimensionless load scaling is recommended for generalizing proven load spectra.

The two most important tractor load spectra are related to the transmission:
- Total *gearbox input spectrum*, see chapter 5.8, Fig. 5.133.
- Total *axle torque load spectrum*, see chapter 5.8, Fig. 5.136.

Load spectra for individual speed ranges of the gearbox need adjustments: Flat for high speeds, bulgy for tillage speeds.

A helpful strategy for "testing" them is the application to proven components which are known not to have reserves.

The relation between class and sum frequency is shown in **Fig. 8.4**. Example for reading the sum frequency: N_1 *is the number of cycles which stands for all loads or stresses of the level* F_1 *and above.*

Class frequencies are necessary for damage calculations (Fig. 8.2).

If the number of classes is high enough, the shape of the sum frequency becomes independent of the class width.

The amount of each column depends on the class width. *A density function requires a constant "class width 1"*. Hence, a continuous line through the tops of the columns is not recommended in general, **Fig. 8.5**.

Lab tests are a must, Table 8.2. Test time is saved first by running 24 hours a day, but in addition by acceleration factors A1–A4, **Table 8.7**. Factor A1 is the most important one, explained in **Fig. 8.6**.

It was found that loads below 50% of the tooth bending endurance limit of a gear wheel cause no damage [8.29].

As the average test load is usually above nominal, local temperatures must be controlled [8.30] and all important data recorded.

8.3 Test equipment for components

A new transaxle is usually the most important subject among all components to be tested (if we exclude the diesel engine).

Three test benches from "low-cost" to "high-tech" are described.

A very simple configuration uses two transaxle prototypes to build up a preloaded power circuit, **Fig. 7.8** [8.26]. Such a test set-up was employed at former MF tests and applied in similar form to testing the transmission Deutz TW 90/TW 900 around 1975. Its investment costs are very low and it requires little energy and cooling equipment as only energy losses are replaced. Its main disadvantages are:

- Two prototypes are needed (costs, time)
- One unit runs in the wrong direction
- Low dynamics
- Four-wheel drive not included

Some disadvantages can be resolved by the test equipment of **Fig. 8.8**. Low-speed hydraulic motors used as pumps create oil flows. Throttling them induces test torques, see equations (7.3) and (7.4) of chapter 7.

Such a structure or similar configurations were used around 1960–70 by Allis-Chalmers. They offer high dynamics and can be completed with 4WD and PTO braking units but convert 100% of the diesel fuel energy to heat. This is expensive and requires large oil-water coolers to remove the heat from the circuits.

Modern transaxle test beds use electric units, **Fig. 8.9**. Braking is done by generators, feeding their energy back into the general grid or building an independent small grid feeding back the braking energy directly to the driving motor. Such systems are very flexible, unfortunately expensive.

Test progress can be visualized by linking actual data of load and cycles with dimensioning diagrams as shown in Fig. 8.3.

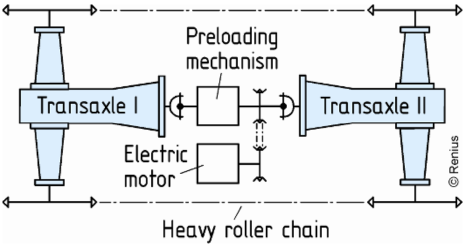


Fig. 8.7: Low-cost equipment for testing two tractor transaxles. Energy input is low but one transaxle is tested in the wrong speed direction

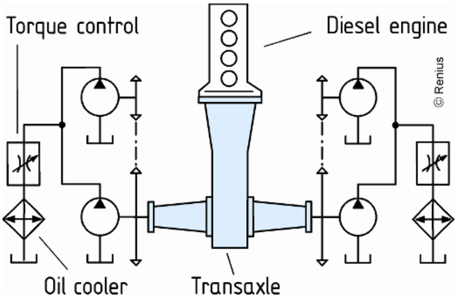


Fig. 8.8: Hydraulic equipment for testing one tractor transaxle using a hydrostatic brake system. Good dynamics but high energy input

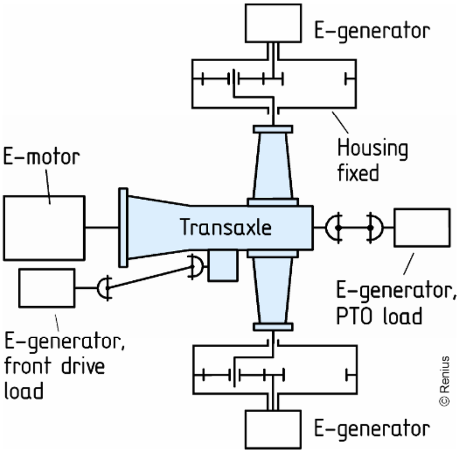


Fig. 8.9: Modern transaxle test equipment with electric units, recuperation of the braking energy, and a highly sophisticated control system

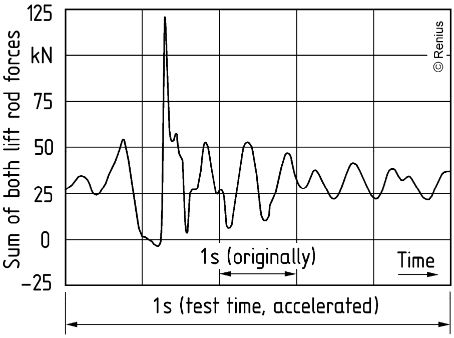


Fig. 8.10: Principle of accelerated test time by increased test frequency in cycles per second. Increased test frequencies enable large acceleration factors (in this case a factor of 5 for the load track of Fig. 7.21): Such tests are possible with the equipment of Fig. 8.11

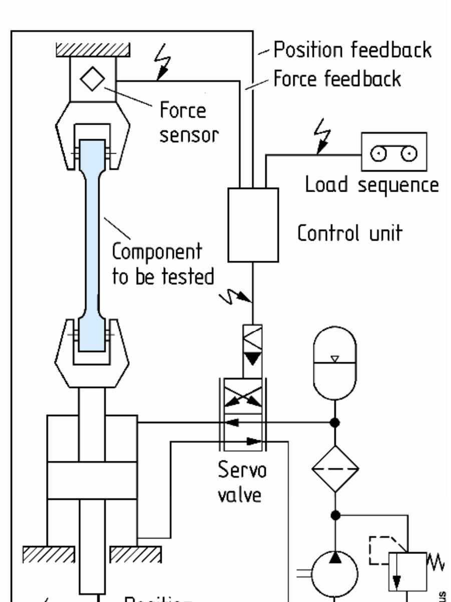


Fig. 8.11: Test equipment for dynamic loads or deflections of constant or random type amplitudes and frequencies. High precision and high frequencies due to closed loop control and special hydraulic components. Commercially available under the Schenck trademark “Hydropuls”

Structures such as axle housings or lift rods of the three-point linkage or steering elements can be tested dynamically with load sequences derived from field measurements. Small amplitudes which may not be relevant to the damage can be eliminated to save time. If the required frequencies are low and the components heavy, simple lab test setups are used with conventional hydraulic cylinder and simple proportional valves for dynamic loading (with or without force feedback). This is, for example, typical for testing a front axle with vertical and horizontal dynamic loads. A higher frequency than that of the original field load track also saves test time, ultimately saving test costs, **Fig. 8.10**.

Smaller components with low deflections under loads such as lift rods or coupling elements for trailer or implements can be tested with considerably increased frequencies. This is possible with test equipment such as in **Fig. 8.11**. A stored load sequence delivers the set value, usually in an automatically repeating form. This signal is forwarded to a servo valve which controls the flow to and hence the load of the hydraulic cylinder. Dynamics and test quality are secured by devices such as:

- Pilot-operated valves for high frequencies
- Accumulators to amplify hydraulic power
- Feedback of the actual load or deflection to the control unit for signal correction
- The use of a special hydro cylinder without friction (hydrostatic centering)
- The use of a high-pressure filter in the line to the servo valve to secure its function

This enables test runs with outstanding dynamics; for example, with up to about 100 Hz in the case of low deflections. The disadvantages are high investment costs and high energy consumption.

Energy consumption can be reduced considerably in special cases such as gear tooth tests with equipment which uses resonance test equipment.

Strain gauges are used to measure material strain, from which stresses or forces can be derived, **Fig. 8.12**. As the cross sections of a strain gauge change slightly, the wire's electric resistance changes accordingly: resistance increases under tension or decreases under compression. This can be utilized by electric *Wheatstone* bridges.

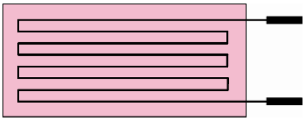


Fig. 8.12: Strain gauge. Horizontal strain reduces the cross section of the wires increasing their electrical resistance and vice versa

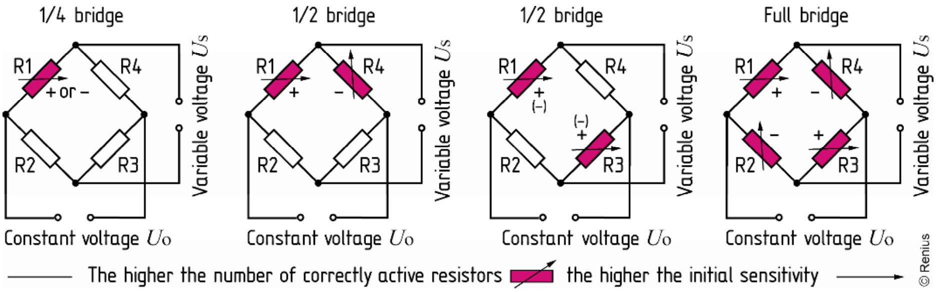


Fig. 8.13: Basic positions of strain gauges (resistors) in a *Wheatstone* bridge to achieve best initial sensitivity. The signs + and – indicate the variation trend of the resistance under strain. Rule for adhering: Opposite resistors should have the same trend of resistance change (both + or both –) while neighbouring resistors, for example R2 and R3, should have opposite trends

Strain gauges are usually produced in foils which are adhered to the cleaned material with high-strength cements [8.31].

Incorporation in a *Wheatstone* bridge requires four resistors and adequate circuitry for good signals [8.32], **Fig. 8.13**.

Fig. 8.14 shows examples for measuring bending moments and tensile forces with strain gauges in full bridges.

A nominal strain of at least 0.05 % (for steels) and compensation of temperatures is required to obtain a suitable electric signal.

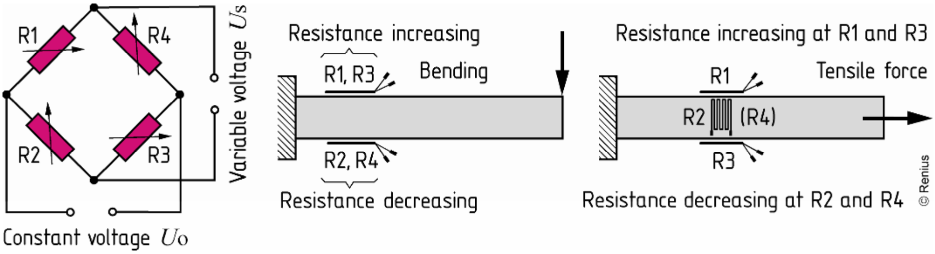
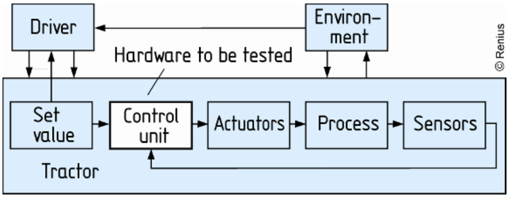


Fig. 8.14: Typical examples of strain gauge applications with full bridges and best possible bridge sensitivities. Sensors with strain gauges should use good steels, enabling high strains.

Left (bending): R1 and R3 (opposites in the bridge) are both fixed on the beam in a longitudinal direction in an area of high tensile stress (strain gauge cross section decreases, resistance increases). R2 and R4 are compressed, resistance decreases at both.
Right (tension): Resistance increases at both R1 and R3 (both opposite in the bridge) while resistance decreases slightly at R2 and R4 due to lateral contraction

The application of electronics has become very important for tractors of upper technology levels, as already addressed in chapter 6.7 (ISOBUS) and 7.2 (automation and safety). Functional complexity has dramatically increased.

Table 8.8: Hardware in the Loop (HiL), applied for testing hardware of an electronic control unit (ECU). The only hardware, in this case, is the control unit. All other system elements are simulated [8.34]. Real-time data flow is a must



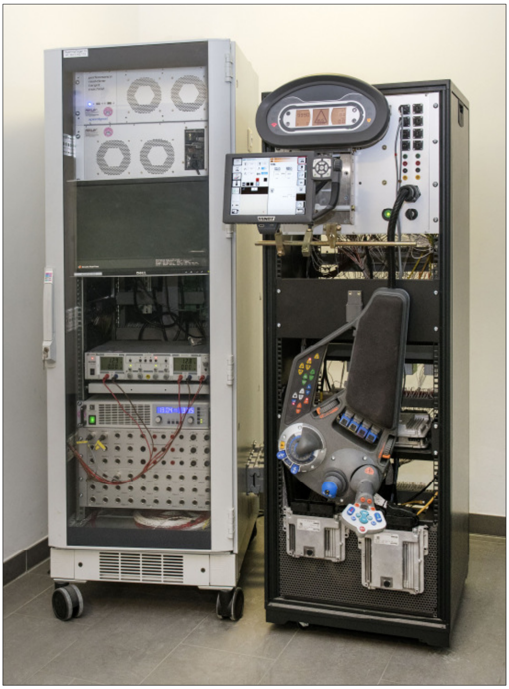


Fig. 8.15: Hardware in the Loop (HiL) equipment for testing hardware components of tractor electronics which are assembled on the right.
The real-time computer on the left simulates the tractor sensor signals. Courtesy AGCO-Fendt

Development is very time-consuming regarding

- Safety
- Reliability
- Functionality
- System dynamics
- System performance
- Ergonomic interactions
- Environmental interactions

Safety, reliability and functionality are must-haves, placing a high degree of responsibility on tractor manufacturers including their engineering staffs. Ergonomics mean easy-to-handle interfaces. Standards and common procedures among different companies support good results.

Systematic development strategies use the V-model as explained in chapter 6 (Fig. 6.61), which includes tests.

Field tests would need a complete prototype, would take much time, and would end up generating unacceptably high development costs.

This means that **lab tests are reasonable** as a first step. In order to save the costs of a complete set of tractor hardware components, **Hardware in the Loop** (HiL) was developed as a powerful tool [8.33, 8.34], **Table 8.8**.

The only hardware in this example is the control unit. Modelling can be supported by commercial software such as Matlab, Simulink, or Stateflow. Supporting test equipment is commercially available working in real time.

Of course, final field tests are necessary with a special focus on “handling” and other ergonomics.

Hardware in the Loop equipment does not need much space, **Fig. 8.15**. The picture shows an application at AGCO-Fendt for testing a single control unit or a combination of electronic components.

The chassis with the rear **axle housing and the three-point hitch linkage** can be tested with one simple configuration, **Fig. 8.16**.

The weight is lifted and released to "fall down" to its lower position where the hydraulic hitch valve closes, decelerating the weight and creating dynamic loads.

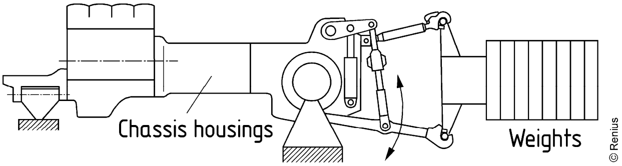


Fig. 8.16: Low-cost dynamic load lab test of the rear three-point hitch and the rear part of the tractor chassis with, for example, 50,000 cycles. Calibration is possible by weights, lift height, hitch valve, and pressure relief valve based on measured load spectra for driving with mounted implements

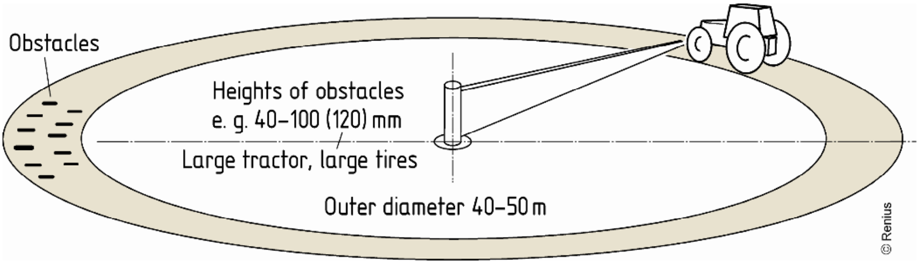


Fig. 8.17: Typical bump test track circuit with obstacles representing a very bad road with potholes. Automatic 24h run of ballasted instrumented chassis (dynamic peak loads) or complete tractor (shaking without implements) for 5–10 days, each representing a total tractor life with a defined probability. Drive by tractor diesel engine or electric pulling unit with reduction gearing. Safety system installed to stop drive in a tractor breakdown or emergency case

Dynamic tests of complete tractors with and without implements are often carried out on track circuits to address tractor durability on bad roads, **Fig. 8.17**.

A chassis or a complete prototype, driven by its diesel engine or an electric drive system, runs on the circle with an automatic guidance system, **Fig. 8.18**.

Two types of test runs are common:

- Ballasted chassis with heavy implements
- Complete tractor without implements

The main challenge is the calibration of the tests, based on real world load or acceleration measurements (load spectra).

Practical road profiles are an important basis for the obstacle system [8.35–8.43]. Fundamentals of track design and its calibration are presented in [8.44].

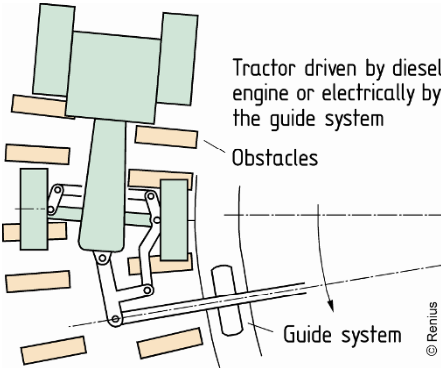

Tractor driven by diesel engine or electrically by the guide system
Obstacles
Guide system
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Fig. 8.18: Mechanics for an automatic guiding system of a tractor which is running on the track circuit. Obstacles are distributed according to road profiles. Calibration by obstacle heights, obstacle correlation (left–right), tractor velocity, tire inflation pressure, and stiffness of axle suspensions

8.4 Public tractor testing

The **Nebraska Tractor Test** is the oldest official tractor test, founded to ensure reliability standards, originally for farmers in Nebraska (Nebraska Tractor Law 1919).

Table 8.9: OECD Tractor Test Codes

No. of OECD Tractor Test Code	
2	Testing of tractor performance
3	Dynamic testing of roll-over protective structures
4	Static testing of roll-over protective structures
5	Measurement of noise at the driving position (driver's ear simulated)
6	Testing of front-mounted protective structures on narrow-track tractors
7	Testing of rear-mounted protective structures on narrow-track tractors
8	Testing of protective structures on tracklaying tractors
9	Testing of protective structures for telehandlers
10	Testing of falling object protective structures on tractors

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Table 8.10: DLG-PowerMix tractor test [8.45]

Group	Operation	
Only pulling	1	Heavy moldboard plowing
	2	Moderate moldboard plowing
	3	Heavy chisel plowing
	4	Moderate chisel plowing
Pulling + PTO work	5	Heavy rotary harrow work
	6	Medium rotary harrow work
	7	Low-power rotary harrowing
	8	Heavy mowing
	9	Moderate PTO work
	10	Low-power mowing
Pull + PTO + hydraul.	11	Distributing dung
	12	Moderate bailing
Pulling trailer	13	Pulling of trailer, mountains
	14	Pulling of trailer, low power

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The *Nebraska Tractor Test Laboratory* (NTTL) is located on the East Campus of the University of Nebraska-Lincoln.

The original building from 1919 was designated as a historic site in 1980 by the American Society of Agricultural Engineers (ASAE). Since 1998 it was housed the *Larsen Tractor Test & Power Museum*.

The John Deere "Waterloo Boy" (25 belt hp) was the subject of the first test, which was completed in 1920 with good results. Test No. 2 was withdrawn and tests 3–7 were carried out with Case tractors.

The main test subjects were drawbar hp (on dry concrete), belt hp, PTO hp, related fuel consumption, and engine oil consumption. Notes on the reliability of the tractor during all test runs were also recorded.

The Nebraska Tractor Test can be regarded as one basis for the OECD test codes prevailing today, **Table 8.9**, for which quite a number of international institutions are now authorized.

In 2019, the *OECD Coordinating Centre* reported the following five most common tests (of a total of 445 tests in 2018):

- 39 % on Code 4
- 23 % on Code 7
- 16 % on Code 10
- 11 % on Code 2
- 8 % on Code 6

Tests on Code 4 are obligatory in several countries, hence dominate.

In order to test and rate the energy efficiency of tractors under *practical conditions*, in 2005 the German DLG TestService launched the DLG-PowerMix, a procedure additional to OECD Code 2.

Based on practical working profiles, the loading sequence of **Table 8.10** was developed: Fuel consumption is expressed in *g/performed kWh for individual tasks and the total sequence*. More information see www.DLG-Test.de/powermix.

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Appendix

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K. Th. Renius, *Fundamentals of Tractor Design,*
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International standards for agricultural tractors

The International Organization for Standardization (ISO) was officially founded on 23 February 1947, starting with standards for tractors (and agricultural machinery) in 1952.

The following list uses shortened titles in order to save space. For complete official titles, search on the respective ISO number online at www.iso.org.

ISO 500-1. Rear PTO (4 basic types), general dimensions, clearances, master shields

ISO 500-2. Rear PTO of narrow track tractors. Special master shield and clearance

ISO 500-3. Rear PTO, shaft locations and shaft dimensions (4 PTO types)

ISO 730. Rear mounted three-point linkage, tractor-related dimensions of all categories

ISO 789. Tests and tractor data measurements,
- Part 1: Power tests for power take-off.
- Part 2: Rear three-point linkage performance
- Part 3: Turning and clearance diameters
- Part 4: Measurement of exhaust smoke
- Part 5: Test of PTOs which are not mechanically driven by the engine
- Part 6: Measurement of center of gravity
- Part 7: Axle power determination
- Part 8: Test of engine air cleaner
- Part 9: Power tests for drawbar
- Part 10: Test of hydraulic power as available for implements
- Part 11: Test of steering capability
- Part 12: Test of low temperature starting
- Part 13: Terms and definitions

ISO 1219. Fluid power, symbols and circuits,
- Part 1: Graphic symbols
- Part 2: Circuits with applied symbols

ISO 1999. Noise-induced hearing loss

ISO 2057. Hydraulic cylinder for implements

ISO 2332. Implement clearance zones of three-point linkage connections

ISO 2534. Road vehicles, gross power test

ISO 2631-1. Evaluation of human exposure to whole body vibration (diagrams)

ISO 3463. Dynamic test of roll-over protective structures (ROPS)

ISO 3600. Operator's manuals

ISO 3767-1/ -2. Symbols for operator control

ISO 3776-1/-2/-3. Seat belts

ISO 4004. Track width

ISO 4251. Tractor tyres,
- Part 1: Tyre dimensions and rims contours
- Part 2: Tyre ply ratings and load ratings
- Part 3: Rims, since 2017 → ISO 18804
- Part 4: Tyre classification code

ISO 4252. Dimensions of cab access and exit

ISO 4253. Seating accommodation, dimensions

ISO 4254-1. Safety, general requirements

ISO 4406. Counting and coding of fluid contamination in hydraulics

ISO 4413. Hydraulics, rules and safety

ISO 4414. Pneumatics, rules and safety

ISO 5007. Operator seat test

ISO 5008. Operator whole body vibration test on a tractor, smooth track and rough track

ISO 5131. Noise level at driver's ear

ISO 5353. Seat Index Point SIP

ISO 5673-1. PTO drive shaft configurations, safety aspects, axial telescopic forces

ISO 5673-2. Drive shaft use and clearances

ISO 5674. PTO safety guards

ISO 5675. Hydraulic couplings, small

ISO 5700. Static test of roll-over protective structures (ROPS)

ISO 5711. Wheel-to-hub fixing, dimensions

ISO 5721. Field of vision. Requirements, tests,
- Part 1: Front vision
- Part 2: Vision rear and to the side

ISO 6336. Calculation of gears,
- Part 1: Basic principles, influence factors
- Part 2: Surface durability (pittings)
- Part 3: Tooth bending strength
- Part 4: Flank fracture load capacity
- Part 5: Materials
- Part 6: Service life under variable load
- Part 7: (under work): Micro pitting

ISO 6489. Mechanical connections,
- Part 1: Dimensions/locations of hitch hooks
- Part 2: Dim./locations of cleves coupling 40
- Part 3: Dim./locations of tractor drawbar
- Part 4: Dim./locations of piton fix
- Part 5: Dim./loc. of non-swivel clevis couplg.

ISO 7241-2. Testing quick action couplings

ISO 7867. Metric agricultural tires and rims,
- Part 1: Dimensions, tire-rim coordination
- Part 2: Load ratings

ISO 8178. Test cycles for IC engines, 10 parts,
- Part 4: S*teady-state* cycles for many applications; cycle C1 includes tractors
- Part 6: Test reports
- Part 10: Test under *transient* conditions

ISO 8434-1. Metallic tube connections

ISO 8608. Road surface profiles

ISO 8664. Radials: load index, speed symbol

ISO 8759. Front mounted implements,
- Part 1: PTO safety and clearance zone
- Part 2: Stationary equipment
- Part 3: PTO: General specifications, location
- Part 4: Three-point linkage

ISO 8935. Cab aperture for implement control

ISO 10448. Hydraulic pressure for implements

ISO 10763. Dimensioning of hydraulic tubes

ISO 10998. Requirements for steering

ISO 11001. Three-point hitch couplers,
- Part 1: U-frame coupler
- Part 2: A-frame coupler
- Part 3: Link coupler
- Part 4: Bar coupler

ISO 11374. Four-point rigid hitch

ISO 11471. Coding of remote hydraulics

ISO 11684. Safety signs and hazard pictorials

ISO 11783. Serial control and communications data network (status 2018),
- Part 1: General definitions
- Part 2: Physical layer
- Part 3: Data link layer
- Part 4: Network layer
- Part 5: Network management
- Part 6: Virtual terminal
- Part 7: Implement messages layer
- Part 8: Power train messages
- Part 9: Tractor ECU
- Part 10: Task controller, information system
- Part 11: Mobile data element data dictionary
- Part 12: Diagnostics services
- Part 13: File server
- Part 14: Sequence control

ISO 12003. Test of front and rear mounted ROPS for narrow-rack tractors, non-rolling,
- Part 1: Test of front mounted ROPS
- Part 2: Test of rear mounted ROPS

ISO 12100:2010. Safety, evaluation of risks

ISO 12188. Test of positioning/guiding systems,
- Part 1: Satellite-based positioning systems
- Part 2: Satellite-based auto-guidance systems

ISO 12368. Test of hook-type connections

ISO DIS 12933. Braking requirements (recalled)

ISO 12934. Definition of agricultural tractor

ISO 13849. Safety of control systems,
- Part 1: General principles of design
- Part 2: Validation of design safety

ISO 14121. Risk assessment of machines

ISO 14269. Operator enclosure environment,
- Part 1: Cab, vocabulary
- Part 2: Cab, climate test
- Part 3: Cab, effect of solar heating
- Part 4: Cab air filter test
- Part 5: Cab, pressurization system test

ISO 14396. Engine power measurement, in addition to ISO 8178 (emissions)

ISO 14982. Electromagnetic compatibility

ISO 15077. Ergonomics for operator controls

ISO 15550. Basics of IC engine measurements

ISO 16028. Large flat hydraulic couplings

ISO 16154. Lighting, signalling, markings for travel on public roadways

ISO 16231. Assessment of tractor stability,
- Part 1: Basic principles
- Part 2: Static stability

ISO 16889. Multi-pass filter test

ISO 17567. Hydraulic power beyond system

ISO 17612. 12V connector, plug in the cab

ISO 18497. Safety of "highly automated"

ISO 18804. Rims, including forestry tractors

ISO 21244. Hitch rings for trailers

ISO 22172-1:2019(E). Repair of IT systems

ISO 23205. Requirements of instructional seats

ISO 23206. Front loader interface for tools

ISO 24347. 80 mm ball-type coupling

ISO 25119. Safety of control systems,
- Part 1: General design principles
- Part 2: Concept phase
- Part 3: Series development
- Part 4: Production, operation, improvement

ISO 26322. Tractor safety,
- Part 1: Standard tractors (including cab boarding steps)
- Part 2: Narrow-track tractors

ISO 27850. Test of falling object protective structures (FOPS)

Comprehension test

The following **135 questions with a few calculation problems** can be used for examinations or personal knowledge tests.

It is also possible (and can be motivating) to jump to the book content using these questions to get a first useful survey on the most important subjects.

The answers can be found in the book on the pages indicated in brackets ( ).

1.1 What are the basic benefits of agricultural mechanization for the general development of a nation (1)?

1.2 What are the benefits of agricultural mechanization for practical farming (1)?

1.3 List six important milestones in tractor history (13).

1.4 Explain the reasons for the success of the Fordson F and its influence on international tractor design (5).

1.5 In which year and in which country were low-pressure pneumatic tires for tractors first introduced (7)?

1.6 How does the Ferguson system work (7)?

1.7 Sketch the design of a modern standard tractor with its main components (15).

1.8 Explain the principle of tractor families using some basic specifications (20).

1.9 Describe the power-related specific initial tractor costs of tractor models within three tractor families. Why does the largest model usually offer the lowest specific costs (20)?

1.10 Explain the basic activities of the R&D, marketing, factory, and finance departments for evaluating a proposed tractor project (20).

1.11 Which are typical criteria when a farmer decides to buy a tractor (21)?

1.12 Which factors affect the ex-factory tractor costs (21)?

2.1 Explain the *Coulomb* equation as a model for soil resistance under shear and normal stress (26).

2.2 Which model can be used to estimate the maximum off-road gross traction (26)?

2.3 Explain the terms “gross traction” and “net traction” and their relationship (28).

2.4 Draw a graph of the gross and net traction of a pneumatic tire versus travel reduction with graphic precision around the zero-slip point for a reasonable zero-slip definition (28).

2.5 Name and explain the definitions for the common dimensionless expression of net traction and rolling resistance (28).

2.6 Which method (with example diagram) is used to plot traction efficiencies without needing slip (30)?

2.7 A large tractor drive wheel is loaded by 20 kN at a specified inflation pressure of 1 bar. Calculate the net traction for 20% slip while driving on an uncultivated stubble field (28).

2.8 Estimate the traction efficiency for the conditions of problem 2.7 (28).

2.9 Explain the benefits of low off-road tire inflation and high on-road inflation (32).

2.10 What is the definition of soil porosity and which value is suitable for successful plant growth (36)?

2.11 Mechanics of a 2WD tractor: Explain how the horizontal pull force results from the rear axle gross traction, using a diagram (41).

2.12 What is a reasonable definition of the locking effect of a drive axle differential (42)?

2.13 Why is the gain in pull (%) by an additional front axle drive much larger than the additionally used front axle load (%) (43)?

3.1 Which are the basic requirements of tractor tires (50)?

3.2 Why is an increased tire width (compared to standard dimensions) more economic in first costs than an increased tire diameter (51)?

3.3 Explain the principle of “low correct” tire inflation for off-road operation (52).

3.4 Dimensions of a tractor may be increased by factor λ. Explain in detail why the dimensions of the tires must be increased by more than λ for off-road operation. Which factor can be recommended (53)?

3.5 What is the No. 1 rule for designing a tractor frame for high torsion stiffness (55)?

3.6 Draw the powertrain of a driven front axle, including a recommended brake position (60).

3.7 Which counting method should be used to achieve load spectra from measurement results of dynamic tractor axle loads (not torques)? How is the absolute highest peak load of the related load history diagram to be read (62)?

3.8 Explain the function of a hydro-pneumatic front axle suspension. Which elements generate the spring effect and the damping (61)?

3.9 Name critical operations regarding the bending and torsion moments of a tractor chassis. What about the role of front implements and front loaders (57, 58, 256)?

3.10 Which is a recommended position for the service brakes of tractor axles with differential gear unit (including bevel gearing) and additional planetary final drive (60, 66)?

3.11 Describe one solution to realize a pull-in-turn function for 4WD standard tractors with Ackermann steering (70, 140).

3.12 Power steering energy supply: Describe alternatives for saving energy by replacing the simple circuit of a constant displacement pump and a constant flow valve (74).

4.1 How does a diesel engine work (81)?

4.2 Which equation indicates efficiency benefits of a diesel engine due to a high compression ratio (81)?

4.3 What is the generally used definition of the "torque rise" (torque back-up) of a diesel engine (formula and diagram) (84)?
What are the pros and cons of high values including the role of initial costs (97)?

4.4 Develop a diagram which contains the complete fuel consumption map g/kWh for a typical diesel engine (94).

4.5 Explain the benefits of a turbocharger with variable turbine geometry (85).

4.6 Explain the function of a common rail injection system of a diesel engine; use hydraulic symbols of ISO 1219-1 (91).

4.7 What is the secret of high needle accelerations of diesel engine injection valves (91)?

4.8 Explain the design principle to compensate for vibrations of the second order of 4-cylinder diesel engines (92).

4.9 Explain the structure of SCR exhaust gas aftertreatment of a diesel engine (93).

4.10 Recommend the fuel tank capacity for a medium-duty 100 kW standard tractor (95).

4.11 Develop a complete energy flow diagram for a heavy working tractor including a realistic utilization factor of the engine working with a manual transmission (100).

5.1 What is the main function of a tractor ground drive transmission and what is its subfunction (104)?

5.2 Explain the configuration of a complete tractor transaxle with front-wheel drive output, PTO powertrain, auxiliary drives, rear axle, and brakes for a modern tractor (104).

5.3 How can a ground drive CVT be controlled to make use of the best possible engine efficiency (engine diagram) (104)?

5.4 Explain the basic principle of saving gear wheels for stepped vehicle transmissions (110).

5.5 Explain a smart method to plot circumferential axle forces versus tractor speed (109).

5.6 Explain the pros and cons of a multiple plate PTO clutch in the rear tractor part (112).

5.7 Which PTO types are standardized in ISO? Describe the power limits of type 1 (114).

5.8 Explain the influence of bending moments on the life of the tractor PTO shaft (115).

5.9 What is the principle and what are the benefits of an "ECO" PTO (diagram) (104, 117)?

5.10 Draw a manual low-cost ground drive transmission which generates 8 forward speeds with only 5 pairs of gear wheels (111, 119).

5.11 How does a "dual clutch" power shift work (simplified sketch) (124)?

5.12 What are the four physical principles of CVT design (with torque generation) (127)?

5.13 Which maximum full-load efficiency can be achieved with a hydrostatic direct CVT gearbox using axial piston units (130)?

5.14 What are the five duties of a charge pump system for a hydrostatic CVT (130)?

5.15 Explain the three basic configurations of an external power split CVT (135).

5.16 How does an "internal" power split CVT work (135)?

5.17 Draw the *Renius* full load efficiency target of a tractor CVT transaxle >100 kW (138).

5.18 Which type of axial piston unit has been proven to offer the best CVT efficiencies (139)?

5.19 Why should a synchronizing shift element be placed near the dominating mass inertia of the transmission (148)?

5.20 Explain *Palmgren's* cumulative damage equation for non-constant loads (153, 262).

5.21 Which level of friction coefficients can be achieved by a journal bearing (155)? What is the main benefit of plotting them versus the term "viscosity x speed / load" (155)?

5.22 What type of oil circuit improves the life of rolling bearings (154)?

5.23 Explain the optimization of the specifications of a pair of gear wheels for a given load spectrum (157, 265).

5.24 What are the two main benefits of shot peening for case-hardened gear wheels (158)?

5.25 Why are gear wheels for vehicle transmissions cut from forged green bodies (160)?

5.26 Draw a planetary gearing (symbols) for summarizing a variable CVT output speed and a constant engine speed. Derive the planetary output speed using the *Kutzbach* method (163).

5.27 Explain the cumulative damage calculation of the *Miner-Haibach* approach (169).

5.28 Which are the two most important load spectra for tractor transmissions (171/172)?

5.29 What is a promising method of generating a transmission efficiency map (173)?

6.1 What are the four groups of operator workloads and what are the main three criteria when evaluating technical aids (180, 181)?

6.2 Explain the closed loop of operating a tractor and the role of its automation (181).

6.3 Which seat adjustments (longitudinally and vertically) can be recommended (184)?

6.4 Plot a simplified (two-dimensional) dynamic tractor model with soft suspensions of front axle, cab, operator, and implement (186).

6.5 Explain the physical conflict between natural tractor frequencies and natural driver-seat frequencies in the cab (187).

6.6 Draw a graph of tire stiffness versus inflation pressure indicating carcass stiffness (188).

6.7 Explain the closed loop control of active pitch damping of mounted implements (189).

6.8 How can an active position control of the operator on the tractor be realized and what are its two main problems (186, 191)?

6.9 What are the basic design principles to reduce tractor noise emissions (192)?

6.10 Which simple design principle is very promising for reducing noise emissions of the engine and gearbox housings (193)?

6.11 Which are the main design rules to achieve a low noise level inside a cab at the operator's ear – at least 8 items (194)?

6.12 Explain the physical function of an air conditioning system (196).

6.13 Draw a graph **e**xplaining the absorbed energy of a solid (full section) steel profile under bending, including high plastic deflecttion. Plot the stress distribution of the cross section for the elastic phase and for the final phase of very high plastic deformation (199).

6.14 What is the principle and loading sequence of testing rollover protection structures meeting OECD Code 4 (200)?

6.15 In which basic areas are electronics applied to tractors (208).

6.16 Explain the basic idea and structure of the tractor ISOBUS (ISO 11783) (208, 210).

6.17 Explain the structure of the V-model for developing mechatronics (210).

7.1 What are the basic practical demands on a tractor-implement combination (table) (217)?

7.2 Explain the function of a closed-loop control circuit (217).

7.3 Safety of mechatronic systems: What are the three basic criteria for classifying the risks of injury or death and for deriving consequences for designing for safety (220)?

7.4 Tractor-implement interface: List the four groups of power transfer with examples (221).

7.5 Electric power supply to implements: Which four electric voltage/power levels are in discussion or even used (221)?

7.6 Describe the two basic configurations of universal drive shafts which generate no additional torque vibrations (224).

7.7 Explain the structure of a wide-angle constant velocity universal joint (drawing) by its kinematic structure (225).

7.8 A lubricated telescoping PTO drive shaft linking tractor and implement is under a torque load of 1,000 Nm. The effective radius of the torque transfer of the telescoping system is 25 mm. Calculate the thrust force for an estimated coefficient of friction (225).

7.9 What are the four benefits of the *Ferguson* system as advertised in its first phase (226)?

7.10 Explain the force system at a rear-mounted plow with draft control using the plow force vectors. What are the benefits (228)?

7.11 How does a "mixed control" system for mounted implements work and what are its main benefits including dynamics (229)?

7.12 A tractor of a low technology level needs a low-cost lower-link sensing for its three-point hitch control. Which mechanically working transducer can be recommended (229)?

7.13 Note six benefits of an electro-hydraulic three-point hitch control system. How can the position control be realized (233)?

7.14 What can be recommended when designing a stable hitch control (4 items) (234)?

7.15 Explain the function of an electric position sensor for a tractor remote valve (234).

7.16 Draw a 4/3 directional proportional valve with electro-hydraulic actuation using the standardized symbols of ISO 1219-1 (237).

7.17 How can a hydraulic motor be driven at a constant pressure system without throttling? Explain the control function using a mathematic model of the torque equilibrium (240).

7.18 Which are the two important disturbing factors of open-center circuits being compensated for in closed center systems, in particular in load-sensing systems (241)?

7.19 Draw the circuit of a common load-sensing system with one variable displacement pump, one consumer, one directional valve, and piping. Use ISO 1219-1 symbols. Explain the benefits of its energy efficiency as compared with that of other circuit types (238).

7.20 Explain the principle of load-sensing systems using a mathematic model of the flow dynamics of the directional valve (241).

7.21 A load-sensing system of a tractor must supply more than one consumer. Which control principle is applied to run the consumers without their influencing one another (241, 243)?

7.22 Explain the "power beyond" hydraulic supply of implements. Main benefit (243)?

7.23 Explain the method of dimensioning safe, fuel-saving hydraulic steel pipes (249).

7.24 What oil contamination can be read from an ISO 4406 code 19/17/13 (252)?

7.25 What are the pros and cons of the two hydraulic circuits – combined or not combined with the transmission oil system (254)?

7.26 A tractor shall be able to carry a front loader. Which are the typical reinforcements? Propose a simple "added frame" design (258).

8.1 What are at least four typical advantages of lab tests versus field tests (261)?

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8.7 Describe the test bench of a dynamic closed loop test of a three-point hitch lift rod (268).

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